THE
ENGINEERING REVIEW.

Edited by
C. EDGAR ALLEN.

Vol. XVIII. MARCH, 1908. No. 3.

Irreversible Engines.

By ROBERT H. SMITH.

SINCE all power-plants are irreversible in the academic sense of this word, it seems superfluous to say anything on the subject of reversibility and irreversibility. The need of doing so arises solely from the persistence with which teachers of thermo-dynamics in books and papers and lectures still exalt "reversibility" as the criterion of the ultimate possible in physical efficiency and economy; and, in particular, adhere to the amazing idea that the "Carnot" cycle, namely, a pair of isothermals crossed by a pair of adiabatics, is *par excellence*, and indeed the one only true, "reversible" cycle for an engine diagram, and is, therefore, the form of diagram towards which all heat-power engineers ought to aspire by way of approximation.

Within the last few weeks has appeared the English translation of Zeuner's great work on "Technical Thermodynamics,"* a work which is a stupendous and compendious record of faithful investigation by the author and innumerable others in the theory of the heat-power use of steam; a work, moreover, which in this, its latest edition, brings its record of physical discovery very nearly up to the date of its publication. Here, in this latest and greatest treatise on the most practical application of the science of heat-mechanics, we find reversibility and irreversibility expounded in precisely the same time-honoured fashion, with precisely the same ever recurring curiously futile and entirely unnecessary declaration that the "Carnot" cycle is the only cycle that is reversible.

Carnot employed this special cycle for the purpose of a theoretical demonstration of a law which is now (although not in his time) called the relation between temperature and entropy. He could not do otherwise because the variables whose relation he was searching for were temperature and entropy or adiabatic function. Every modern student must use the same cycle when he is studying this same relation. If he be studying the relation between

* "Technical Thermodynamics," by Dr. G. Zeuner, translated by J. F. Klein. 2 Vols. London: Constable & Co.

temperature and volume he must necessarily make use of a cycle-diagram made by two isothermals crossing two isometrics: if the subject of study be the direct relation between temperature and pressure, the diagram used must be a cycle of two isothermals crossed by two isobarics.

In order to reduce his argument to the simplest possible elements, Carnot imposed upon the thermodynamic action round this cycle certain restrictive conditions. For instance, one of these was expansion and contraction carried out so quietly and slowly that the kinetic energy generated and destroyed should be minutely small and, therefore, negligible in the calculation. This extreme slowness of action enabled him to specify another restriction which he desired, namely, minutely small temperature differences between the "working substance" and his heating agent from which this took heat by conduction, and again between the same and his cooling agent which abstracted heat by conduction. These two restrictions involve a third condition, namely, close approximation to uniformity of pressure and temperature condition throughout the whole mass of the "working substance." All these three conditions were approximative only; the cycle action could not take place at all without the real effective existence of the small differences, but they were specified to be so small that certain effects due to them which were irrelevant to the problem in hand were eliminated and the problem thus simplified.

Now throughout the history of thermodynamics since Carnot's time, at least so far as it is recorded in published books, the enormous mistake has been made of thinking that these restrictive conditions were essential to his argument. Carnot declared that the thermodynamic action round his cycle was "reversible." This again was only an approximation: for instance, the negligibly small temperature-differences must be reversed in sign in order to carry round the cycle in reverse direction, and this change in the temperatures is the measure of one of the two deviations from strict reversibility. Apparently all subsequent writers have fallen into the strange error of thinking that the above simplifications of the problem constituted the conditions of reversibility, and that without them there must infallibly be irreversibility.

Proofs that this is so can be culled from nearly every book on the subject. Zeuner, who is always explicit and sometimes almost desperately so, explains (p. 28, vol. i.) that "the body-tension," which would be better termed "body-stress," or "the pressure exerted by the body from within against the external envelope" and "the pressure by the envelope against the body's surface" must be equal, and that when they are unequal, irreversibility results. How, in the name of common sense, can these be ever under any circumstances unequal? Does not Zeuner believe in the fundamental axioms of mechanics as formulated by Newton? This sort of mischief arises solely from the mistaken notion that kinetic energy generated during expansion is developed by "internal" work done by the accelerated mass itself. This is a grave error. The work of generating or destroying kinetic energy in each small element of the expanding mass is done not by that same element itself but by the element contiguous to it on the path of acceleration. In an explosion or violent outflow of gas there is an extremely rapid drop of pressure from the heart of the exploding mass to its outside "envelope," but at no instant of the explosion is there at this mathematical "envelope" a difference of pressure of a hair's-breadth between the two sides of this mathematical surface.

The validity of the Carnot argument does not depend in the very least degree upon the restrictive conditions which are supposed to make his cycle the one only true reversible cycle. They only make the problem easier to think out. The truth of the propositions remains exactly the same, however fast the expansion may be, and whatever differences of temperature there may be between furnace-gases, steam, and condenser. The

isothermal-adiabatic cycle is in essence no more reversible than any other. Every sound student of mechanics knows that no sudden drop of pressure or other stress ever exists for an instant across any mathematical dividing surface between any two portions of matter. He knows that all accelerations and kinetic-energy generations are effected by pressure-gradients, which may be gradual or steep but which never break suddenly. Every sound student of heat knows that conduction of heat never takes place by virtue of any finite temperature difference across a mathematical dividing surface, but always by a temperature gradient. The rate of supply or abstraction of heat is proportional always, not to a temperature difference merely, but to a temperature gradient.

In a steam-engine it makes no difference in the thermodynamics of the steam expansion whether the work it does on the piston is spent in producing kinetic energy in heavy pump spears and beams or is spent otherwise without any generation of accelerated motion. Similarly in the rapid outflow of steam from a safety valve it makes no difference in the thermodynamics of the work done by each minute expanding particle, whether this work is spent in increasing the kinetic energy of the steam in front of it along the stream line of outflow or is spent otherwise.

As to irreversibility, every engineer knows that a rapid accelerated outflow of fluid is reversible, because examples of it may be seen all over the world every hour of every day in a million machines, such as injectors, ejectors, fans, &c. It is as reversible, in the strictest and fullest and most academic sense of the term, as is the kinetic energy of an upthrown stone reconvertible into gravitational potential energy. Neither the uniformity of pressure throughout the expanding mass, nor the quietude and slowness of its expansion, have anything to do with reversibility and irreversibility.

The sole element in the action that is irreversible is one which must always necessarily be present so long as there is any thermal action at all: it cannot be avoided, because it is fundamentally essential to the passage of heat. It is simply that heat transmission, whether by conduction or radiation, is always in the direction of the down gradient of temperature; heat cannot be conducted or radiated up against the temperature gradient. Indeed Zeuner, after forty pages of learned disquisition and calculation upon reversible and irreversible functions, lets fall quite incidentally (see page 83, vol. i.) the tacit admission that irreversibility consists essentially in "the direct passing of heat from a body of higher to one of lower temperature, to *heat conduction*." Unhappily this slip into plain common sense comes within half a page of the end of the long chapter devoted to the subject.

It has quite often been stated by various physicists that the real meaning of the second law of thermodynamics, which is established by the Carnot reasoning, and on which is based the doctrine of dissipation of energy, is neither more nor less than that heat travels by conduction along the down gradient of temperature. Unfortunately on the top of this there has been deposited a lot of mathematical futilities and irrelevancies which have come to be thought part and parcel of this simple law. It is as if, in demonstrating the dimension-relations between a sphere and a cone necessary to make their volumes equal, we assumed that they were both of the same colour in order that our minds might not be distracted by irrelevant differences of appearance; and then our scientific successors, after learning this lesson, habitually took for granted that a sphere and a cone could not be equal in volume unless they be of the same colour. Or as if, when studying the elementary laws of heat conduction, we considered, for the sake of simplicity, conduction through material of uniform conductivity, and then, being weary of finding out laws, abided through the rest of our lives in the comforting faith that if the conductivity be not uniform, then no conduction at all is possible.

Take any random curve of expansion. The thermodynamic action along it can be analysed by resolving the curve into a great number of minute steps alternately isothermal and adiabatic. This analysis is frequently employed in thermodynamic treatises, and its legitimacy is universally acknowledged. According to current notions of reversibility, each minute step would be a "reversible operation." If the elements of the curve be reversible, how can the curve itself as a whole be otherwise than also reversible? At one time it is said that isothermal and adiabatic curves are the only expansions that are reversible: at other times it is explained that quietness of expansion and uniformity of temperature condition through the expanding mass are the requisite conditions of reversibility. How can such statements be possibly reconciled? Unquiet expansion is quite as readily possible along these two curves as along any others. It so happens that unquiet expansions almost always approximately follow adiabatic curves; and this fact is quite commonly known. The only frequent exception is that of explosions, in which heating by chemical combustion accompanies expansion. The same expansion curve can be also mathematically analysed into minute isothermal and isobaric steps; or into isothermal and isometric steps; or into isobaric and isometric steps. The varieties of such possible mathematical analysis are innumerable. All are equally legitimate; the four above mentioned and several others are all equally useful.

This question is not one merely of theoretic import. It is of enormous practical consequence to have clear ideas upon it, because what is so hazily defined as "reversibility" is also rightly declared to be and logically proved to be the criterion of *perfect physical efficiency*. This means that in a reversible operation there is *no dissipation of energy*. It is, therefore, of fundamental importance to practical engineers.

Now let us look closer into this question of thermal reversibility. In doing so we will discover a most significant difference between the laws of heat and of mechanical pressure, a difference that strongly militates against Dr. Zeuner's analogies between gravity and heat potentials which led him to give the name "heat-weight" to what is now universally called entropy.

Heat flows by conduction down the temperature-gradient. If through any thickness of material the heat conductivity and the temperature-gradient be uniform, then as much heat flows out at one side of the thickness as enters at the other: there is no rise or fall of temperature or of pressure anywhere in this thickness due to the conduction; there is no expansion or contraction due to the flow of heat: and the material occupying this thickness does no work because of the flow. If the temperature-gradient be not uniform, or if with uniform temperature-gradient the heat conductivity be different in different parts, then the flow of heat down the temperature-gradient results either in an accumulation or in a conductive abstraction of heat; accumulation taking place in those parts where the down-temperature gradient is less steep or where the heat-conductivity is less, following parts where these are greater; that is, the accumulation depends on the second-gradient of temperature and is positive where this second-gradient is positive, the first-gradient being negative. It is this *change* of temperature-gradient or of conductive capacity that is the essential condition of any supply or abstraction of heat by conduction. Such heat supply, in a manner regulated by external mechanical conditions, will develop expansion or other strain, and in so doing leads to mechanical work being done by the heated body. The limit of the external regulation is the prevention of all straining, when the external work becomes zero and instead of it storage of elastic potential energy occurs. Thus a *uniform* down-grade flow of heat develops no capacity for doing work in the materia through which it flows, and the drop of heat energy from higher to lower temperature is entirely wasted in so far as regards generation of mechanical

power. It is this *uniform* down-flow of heat which results in dissipation of heat energy. This is not a suitable place in which to go further than to point out that, while dissipation of heat energy depends on the first-gradient, it is the *second-space gradient of temperature* to which is due the possibility of spending heat either in doing external mechanical work directly or in storing up elastic potential energy which may subsequently be spent in doing such external mechanical work. Now compare these functions of the temperature-gradient with those of the similar gradient of mechanical pressure. In order that material through which mechanical pressure is exerted may not absorb kinetic energy from the pressure-transmission, the pressure itself must be uniform through the whole thickness of the material. Kinetic energy is generated or destroyed only when the pressure is not uniform, that is, when the pressure gradient is not zero. Thus uniform pressure in regard to kinetic energy generation corresponds strictly to uniform temperature-gradient in respect of accumulation of heat; and a non-zero first-gradient of pressure corresponds to a non-zero second-gradient of temperature. If one wished to follow this correspondence to the point of searching for the mechanical analogy to temperature, it would be found to be the product of a stress by a length, which is quite a different thing from that of a force by a length.

To protect oneself from criticism it must here be noted that parallel fluxes of stress and of heat alone are spoken of above; if either flux spread laterally by variation of cross-section of the material, then the wording of the propositions needs to be slightly changed to take such spread into account.

On the other hand it will be found that the potential energy of elastic strain can be accumulated under zero stress gradient. Indeed zero stress-gradient is the condition under which all external work done by or upon straining material is wholly derived from or spent upon elastic potential energy and none of it from or upon kinetic energy.

Now there are two modes by which temperature and store of heat can be changed. The one is by conduction or radiation of heat; the other is by adiabatic straining, *i.e.*, expansion or compression, &c., the word "adiabatic" meaning "without heat conduction or radiation." Such straining is strictly "reversible" in the academic no less than in the practical sense of the term.

In thermodynamics, therefore, all change in temperature and in "heat-content" accomplished by heat conduction or radiation—and one ought here to add by electric current conduction—is irreversible and involves dissipation of energy; while all change in temperature and "heat-content" effected by adiabatic straining is reversible and involves no dissipation of energy. The latter case is merely the limit of the former when the conduction and dissipation both become concurrently zero.

Testing a Centrifugal Fan.

*IV.**

By C. L. BROWNE.

(*Continued from page* 380, *Vol.* 17.)

Now the actual blade angle at intake will be considerably less than that of mean flow, and an investigation into the ratios $\frac{c_2}{u_2}$ in the vast majority of fans when working at their normal orifices will show that in order to make cot. $\theta = \frac{c_2}{u_2}$ the angle θ will have to be comparatively very small.

We are consequently met with the practical difficulty that in trying to obtain what would appear to be the theoretically correct blade angle at intake, we shall be seriously curtailing the area of the radial passage through the wheel. This will be made evident by consideration of Fig. 9, and by

* The whole of this article should be read before the paragraph commencing "This efficiency of the fan alone" in the right hand column of page 376, December issue.

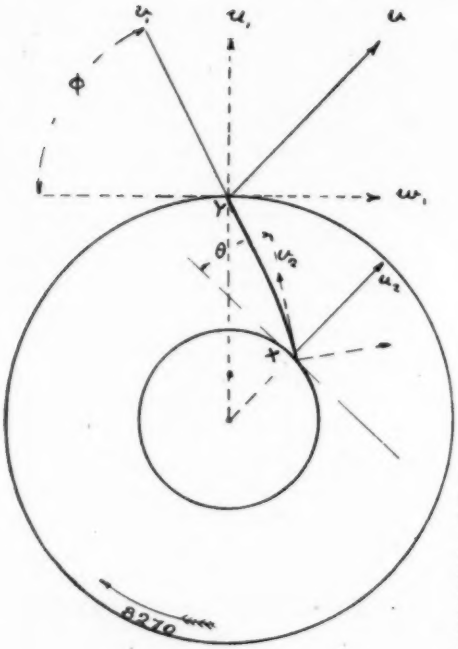


FIG. 7

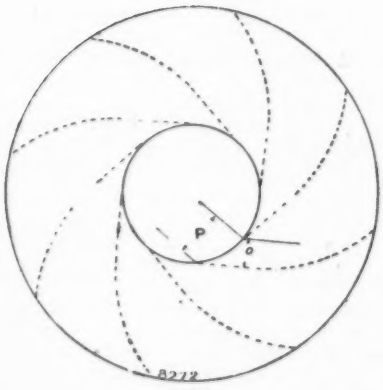


FIG. 9.

noting the difference between the dimensions *P* and *O*.

In order to prevent loss of head from enlargement of the radial passage through the wheel, it is usual to make the width of the blades at outflow less than at intake, and this involves the making of the sides, or shrouding, of the wheel conical.

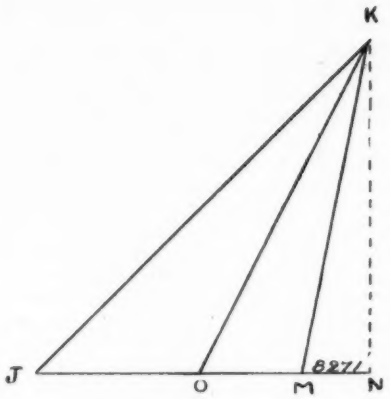


FIG. 8.

Thus a little consideration will show that if the blade deviates in any way from the radial, the angle made at the joint between the blade and the shrouding will vary considerably along the radial length of the blade. As this joint is usually made, either by flanging back the blade itself, or by riveting to it a length of angle iron, it is evident that the production of a varying angle at the joint between the blade and the wheel shrouding will be a comparatively expensive operation, and one unlikely to appeal to fan manufacturers.

Referring again to Fig. 7, v represents the actual velocity and direction of a particle of air as discharged from the fan wheel into the volute formed by the fan housing, which can be split up into its radial and tangential component u_1, and w_1, respectively, w_1 being very nearly the path taken by the air in passing through the volute formed, by the fan housing.

The angle made by the line v_1 with the tangent to the circle described by the outer tips of the blades is designated by the letter ϕ on the diagram. It should be noted that the angle ϕ on the diagram is *not* the tip angle of the fan blade, but that of the mean flow of the air, which differs from the former by from 15 degs. to 40 degs., the amount of this difference depending upon the length and number of the blades and upon the degree of their curvature.

When it is borne in mind that the water gauge set up by the fan at constant orifice varies approximately as the square root of

the velocity of whirl, and that w_1 may differ considerably from c_1, the tip speed of the blades, it is evident that the manometric, or pressure efficiency of the fan at constant tip speed and orifice will be considerably affected by varying the blade angle at tip, and consequently the angle ϕ.

Following the notation used it will be evident that $w_1 = c_1 - u_1 \cot. \phi$, and as ϕ is always measured clockwise, if ϕ be greater than 90 degs. cot. ϕ will be negative, and w_1, which approximates to the velocity of whirl, will be greater than c_1, the tip speed of the wheel. A practical example of this will be given later.

The particle of air is now assumed to be leaving the wheel with a tangential velocity w_1, and if the orifice of the fan were of such a size as to allow the air to be discharged at this velocity, the whole of its kinetic energy $\frac{w_1^2}{2g}$ would be lost. For this reason a drop in velocity is allowed to take place in the volute, and the discharge area is designed accordingly.

Referring again to Fig. 8 it was shown that in order for a change of velocity and direction to take place from *J. K.* to *J. M.*, the least loss of kinetic energy took place when the final velocity was *J. N.*, the loss in kinetic energy being $\frac{K N^2}{2g}$.

In this case, however, it is required to change the velocity w_1 to some other velocity of discharge such as will give the maximum gain in pressure, and it can be easily proved that this will be the case when the final velocity *J. O.* = *O. N.* Since the angle between the direction w_1 and that of discharge through the volute is small, the discharge velocity v_4 will be $\frac{1}{2} w_1$.

Now, as pointed out previously, in the majority of commercial fans, the radial velocity u_1 is by no means uniform over the discharge area of the wheel, and it is generally difficult to assign a correct value to ϕ, and consequently the value of w_1 and hence the value of v_4 can only be approximated.

There are a number of highly efficient fans, such as the well known "Rateau" fan manufactured on the continent, which have been most carefully designed on theoretical lines, but which would be far too costly in production to compete with the steel plate fans offered in this country for general purposes by American and English firms, the proportions of most of which have been settled empirically with a view to putting on the market a fan of simple construction and with moderate manometric, volumetric and mechanical efficiencies.

From what has been previously stated it will be evident that a fan with a high mechanical efficiency will be a large fan, which means that its weight and consequently its cost per h.p. in the air discharged will be high. The design of a centrifugal fan would be considerably modified by the use of a properly proportioned expanding chimney fitted to the fan discharge, in which a portion of the velocity head in the air discharged is changed into pressure head.

Both the manometric and mechanical efficiencies would be considerably improved. These chimneys are frequently used on the continent with a great saving in the power required to drive the fan, but the short nozzles which may be sometimes seen on installations in this country are chimneys only in name.

Not only is the first cost of a properly designed expanding chimney an item worth considering, but its employment, of course, involves a higher water gauge for a given tip speed, and where the fan is direct driven, a slower running and consequently more expensive motor or engine will be required, but any expenditure in this direction would be capital very well invested.

Fan Efficiencies. It is usual to express the work done by a fan as the volume of air in cub. ft. per minute delivered against a head measured in inches of water. As the head corresponding to a column of water an inch high is equivalent to a pressure of 5·196 lbs. per sq. ft., if h = the head in inches of water and Q the cubic feet of air delivered per minute against that head, the ft. lbs. of useful work performed per minute by the fan becomes $5{\cdot}196\ h \times Q$, and the air h.p. $\frac{5{\cdot}196\, h \times Q}{33{,}000} = \frac{h\,Q}{6{,}352}$. If the b.h.p. delivered to the fan wheel is taken to be the h.p. absorbed by the fan, the h.p. in the air as expressed above divided by this value will be the mechanical efficiency of the fan.

(To be concluded.)

Recent Gas Engine Research.

II.

(*Continued from page* 104.)

From the set of indicator cards of Fig. 2 it appears that the ignition is sometimes spontaneous, or starts from some point on the compression line before the ignitor has acted, indicating pre-ignition of the charge. A still different sort of variation or lack of constancy is shown in Fig. 3; the variations are those due to height of combustion lines and the form of that part of the combustion line which runs into the expansion line. In Fig. 4 there is presented another indicator card in which not only is there a variation of the combustion line similar to that of Fig. 3, but one of another sort; a violent wave appears at times, different for successive strokes and sometimes absent.

An examination of the indicator cards here presented and many others of a similar kind that every gas engine experimenter has found at some time or other will indicate that the variations in the combustion line are principally of three sorts, running one into the other; first, there may be too early a beginning of the combustion line or pre-ignition, which comes and goes sometimes in the most puzzling fashion, but which at other times can be traced to a removable cause and eliminated; secondly, with an absolute constant ignition and smooth lines successive strokes may indicate a displacement of whole or part of the combustion line. This is a mixture variation effect. Thirdly, there may be at some time violent waves or even mild waves differing on successive strokes, passing away and recurring at times, and at other times persistently present. This is the phenomenon of the explosive wave.

Mixture Effects.—A variation of mixture may affect the combustion line through a change in the rate of propagation, which results from changes in mixture proportion considered in conjunction with piston speed. A slow burning mixture will tend to give a flatter combustion line with a fixed piston speed than a fast mixture. Likewise, a mixture may begin to burn rapidly and finish slowly, giving succeeding combustion lines which coincide in part, but which vary toward the end where the combustion line runs into the expansion line from the dilution of the last part of the charge by early produced neutral gases. Through excessive dilution of some part of the mixture in the cylinder, which it must be understood is probably not homogeneous, some of the gas may not burn, and on succeeding strokes the diffusion may be more or less complete than before, allowing the incompleteness of the combustion to vary toward the end of the process. The actual mixture under combustion consists not merely of air and gas, but rather air, gas and burnt or neutral gases. Any variation of proportion of the quantity of air, gas or burnt gases to the whole that may occur will produce variations in combustion lines, but variations in combustion lines may just as well occur when the proportions of totals are constant, through lack of homogeneity of the mixture on successive strokes.

Excluding for a moment that consideration of neutral products the problem of securing a proper proportion of air to gas in the cylinder is one of orifice flow, and the failure to secure it may be analysed on the basis of the laws covering orifice flow. In this connection it must be remembered that it is not volume proportions that are most important, but rather weight proportions, since it is a definite weight of air that is required to burn a definite weight of gas, although volume proportionality will follow if the pressure and temperature of both the air and the gas are constant and the same, which unfortunately is seldom true. The orifices through which the air and gas flow separately to form the mixture are of very peculiar forms, as a rule, and not the same either in size or form, so that the laws of variation of proportion are reducible to the laws of variation in the weight of air per pound of gas flowing through separate orifices of different form and size at probably different temperatures and with different pressure drops or pressure heads.

It is well known that the co-efficient of efflux for the flow of gases through orifices varies with the size of opening, shape of

opening, and velocity of flow or pressure head. Air enters the engine cylinder under the influence of a pressure head represented by the cylinder vacuum. The gas, however, has a pressure higher than atmosphere if pressure gas and lower than atmosphere if suction producer gas, so that while the head causing the flow of air is the cylinder vacuum alone, the head causing the flow of gas when under pressures is the sum of the cylinder vacuum head and its own pressure head, and when under suction is the difference between the cylinder vacuum and the gas pipe vacuum. Gas pressures are, moreover, never constant in practice, nor will any of the gas pressure regulators proposed and used make them constant nor reduce them uniformly to atmosphere because they always involve inertia effects of moving solid parts and of the gas itself.

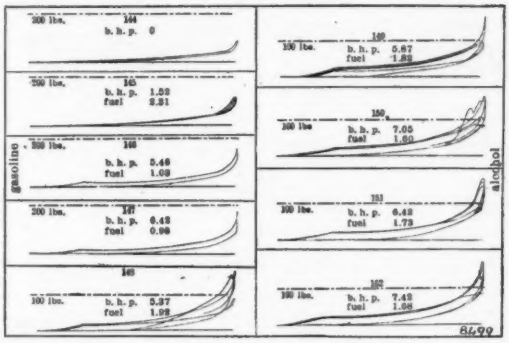


FIG. 2.—SHOWING PRE-IGNITION EFFECTS.

With a fixed opening the cylinder vacuum head acting on the orifices is a variable, because piston speed in engines varies from zero to a maximum and back to zero for every suction stroke. This variable vacuum head with fixed gas pressure head either positive or negative causes a variation in the ratio of the total head on the gas orifice to the total head on the air orifice, and hence has the effect on varying proportions of air to gas throughout the stroke. In addition to this, whenever the gas is under pressure, and the opening fixed, the idle engine period, that is, the period of no suction, allows time for the pressure gas to flow past the orifice and collect in the air chamber, which would tend to make the mixture rich in gas at the beginning of the next stroke.

Intermittence of flow is another element which enters into the variation of proportion because it brings into play the inertia of the gas and the inertia of the air. A stream of air or gas cannot be started or stopped instantly, and as the masses to be moved are not the same the inertia will not be the same for the two, and one will tend thereby to lead in its flow over the other one that which has the smaller mass leading. At the end of suction that which has the greater mass will continue its motion for the longer time.

It appears, therefore, to be an extremely difficult proposition, viewed entirely independent of the gas engine, to secure constant weight proportion between two gases flowing through two orifices into a partial vacuum through openings of different sizes and shape under heads compounded of the vacuum and the gas pressure with variable rates of flow, changes of barometer, gas pressure, and the temperature of both gases, and it is not surprising that variations occur, but rather more surprising that the results are as uniform as they are. After having proportioned the air and gas, the mechanism delivers it into a cylinder through a valve to the irregular head or clearance space, where it mixes more or less uniformly with the neutral gases therein. These residual gases may have the same composition on successive strokes or may not, depending upon a variety of circumstances, some uncontrollable, such as

FIG. 3.—VARIATION IN HEIGHT OF COMBUSTION LINE.

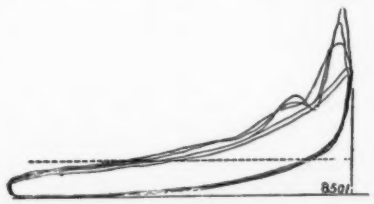


FIG. 4.—SHOWING VIOLENT WAVES EFFECT ON INDICATOR CARD.

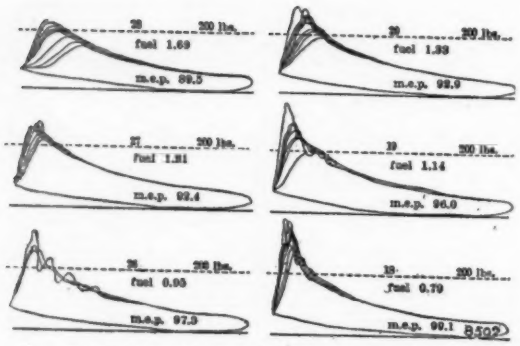


FIG. 5.—SHOWING EFFECT OF VARYING THE MIXTURE.

diffusion, others under practical control, such as point of ignition and back pressure.

A rather aggravated case of this mixture variation effect on various engine settings is given in Fig. 5, the fuel being gasoline and the engine a small one.

Another case not so aggravated, but much more common, is shown in Fig. 6. This is from a medium size engine running on city gas. It is to be distinctly understood that the mixture variations which occur in a hit-and-miss governed engine before and after misses when the combustion chamber contains in the one place air after a miss and in another case burnt gases after an explosion, are excluded from this discussion, and only the variations which occur in engines operating under steady and uniform conditions included.

Explosive Waves.—The French scientist Berthelot gave the name "explosive wave" to a certain phenomenon observed in the combustion of explosive mixtures. This phenomenon may easily occur and does occur in gas engine cylinders. In some cases it is possible to define the conditions which will produce it and in other cases it is not. Examining the rate of propagation through a tube, it is found that at times the propagation is uniform, at times mildly undulatory, indicated by waves of small amplitude, and at times violently undulatory, indicated by waves of great amplitude accompanied by shock and sound. This violently undulatory propagation has an extremely high rate and can be produced whenever there is a violent agitation of the mixture about to be ignited.

One source of agitation producing this result, and in use by early experimenters, was a small stream of the mixture impinging into the main mass. An apparently different agitation, though probably identical, studied especially by later experimenters, is a pressure wave or compression wave. It can be shown that if combustion be started in a tube, closed at one end, waves may set up so violent as to cause extinction before the passage through the tube is completed. In this case the agitation is a result of a compression wave produced by the combustion itself. In engine cylinders this same sort of wave may exist. The motion of the piston itself during compression produces a compression wave which advances before it through the mixture and which probably reflects and superimposes or neutralises, as accident may dictate, so that the entire mass is in a process of agitation during compression.

Inflammation started in such a mixture, agitated either by streams of gas as the result of pockets in the combustion chamber or by compression waves, will sometimes be very violent, giving a true explosive wave, but may not exist at all. This seems to indicate that the violent momentary pressures of the explosive wave crest result only when advancing waves superimpose one on another and synchronise with their reflections.

A simple experiment that can be performed by anyone will yield explosive waves of this sort on any gas engine if between the indicator and the engine cylinder there be connected a pocket with a small throat, which may be made of pipe fittings. An engine which gives a perfectly smooth combustion line without such a pocket will give with the pocket explosive waves even when the ignition is quite late. In nearly every engine these waves will be produced when the ignition takes place before dead centre, that is, during the time when the mass is agitated by compression waves from the piston.

FIG. 6.—MIXTURE VARIATION EFFECT ON ENGINE RUNNING ON TOWN GAS

Fig. 7 shows an explosive wave from a kerosene engine having no hot bulb and a

very good form of combustion chamber. Fig. 8 shows for even very late ignition an extremely violent wave, so violent that it is shown for only part of the stroke, the pencil having jumped from the paper for the rest of the stroke. Fig. 9 is a record of a violent wave in a large oil engine of the hot bulb class, and Fig. 10 shows another less violent wave taken from the same engine shortly after. These pressure waves are not to be confused with the occasional fluctuations of the indicator pencil due to the natural period of vibration of the piston and parallel motion of the indicator, although, according to my experience, the confusion is more likely to be the other way, the vibration of the indicator parts being more often the only explanation for the waves that are found.

Pre-ignition.—Whenever on compression a mixture ignites itself before dead centre the phenomenon is called "pre-ignition." Besides the many known easily avoidable causes, there are some that are difficult to understand. Any inward projecting part, such as a piece of asbestos gasket or rough edge of the casting, a bolt head, nut, piston compression plate, carbonised oil or possibly an ignitor, may get so overheated as to cause ignition. The compression causes a temperature increase measured by the degree of the compression so that all parts of the gaseous mixture, except those directly in contact with cold walls, will suffer the same temperature rise, due to the compression. If there is near any particle of mixture another source of heat than the compression, the temperature at that place will rise higher and may rise so much higher as to cause an ignition. It may be also that lack of homogeneity in the mixture will result in zones where the mixture has a lower temperature of ignition than at other places: for example, in places where lubricating oil is vaporising, or in the case of gasolene, where the mixture is a little more rich in gasolene. This is another cause. In spite, however, of these traceable causes there seem to be some others, and these are mostly associated with the percentage of hydrogen in the gas.

At one time it was believed that the temperature of ignition of hydrogen was so low that the addition of hydrogen to a gas not previously containing it would lower the temperature of ignition of the mass, and designers, including the writer, went so far as to announce figures for the reduction of compression for each per cent. of hydrogen present that was necessary to prevent pre-ignitions.

Repeated experiments by the different

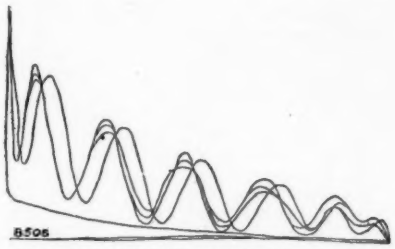


FIG. 7.—EXPLOSIVE WAVE FROM OIL ENGINE HAVING NO HOT BULB.

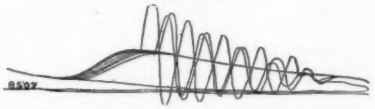


FIG. 8.—AN EXTREMELY VIOLENT WAVE.

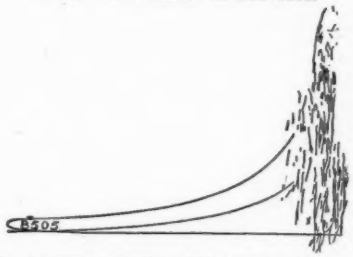


FIG. 9.—EXPLOSIVE WAVE FROM LARGE OIL ENGINE.

FIG. 10.—SOMEWHAT LESS VIOLENT WAVE.

engine builders and by engineers not associated with the building of engines point conclusively to the fact that pre-ignition may occur when the percentage of hydrogen is low and may not occur when it is high, and again may occur when it is high and may not when it is low, for a given engine running on a given compression; but there seems to be substantial agreement on the statement that if the hydrogen were absent there would be no pre-ignition at this compression.

A considerably detailed investigation carried on partly in the laboratory at Columbia and partly in the field seemed to indicate that it was not the percentage of hydrogen in the gas that fixed the tendency to pre-ignite, but rather some ratio of the hydrogen to the other elements present. The remedies applied commonly for

pre-ignition troubles are two-fold: first, a reduction of compression; second, an introduction of neutral elements, such as water to be vaporised into steam, steam itself or cooled and purified exhaust gases. This practice introduces greater variations in the mixture than it is desirable to have, and is justified only in emergency, that the engine may continue to run.

An examination of the old values of the temperature of ignition for explosive mixtures throws no light whatever on the solution of this phenomenon, but some more recent determinations do. By dropping a weight on a plunger, closely fitting a cylinder containing a known gas mixture and measuring the plunger travel up to the point where it is stopped by the explosion, the temperature of ignition may be calculated by the adiabatic law from the volume ratio in compression when the ignition takes place by the compression alone. This method of measurement was employed by Dr. K. G. Falk in the mechanical engineering laboratories of Columbia, and gave for repeated trials of the same mixture at different times the results so consistently uniform that they seem to be very valuable. These were reported in the October meeting of the American Chemical Society.

The results obtained explain the apparent inconsistency between percentage of hydrogen in the gas and the conditions of pre-ignition. It appears from the figures given for the temperature of ignition that in a producer gas containing hydrogen and *CO* with various neutrals mixed with oxygen the temperature of ignition does not depend on either the hydrogen necessarily nor the *CO* necessarily in the mixture, but on the relation that one of these bears to the oxygen present and which one can be determined only by computing the temperatures of ignition for the value and taking that value which is lower. One very significant fact in addition to the above is that the ignition temperatures and compressions formed are all very much higher than those used in engines. No ordinary engine uses compressions anywhere near those determined for pre-ignition.

It is evident, therefore, that as pre-ignitions occur they are due not only to the compression, but also to other sources of heat. The interior parts must be hot enough in places to materially augment the temperatures produced by compression alone. As the final temperature, due to compression, bears a fixed relation to the initial temperature for any given compression, that final temperature may be made higher not only by heat additions during the compression, but by a higher initial temperature. High temperature burnt gases retained in the cylinder are, therefore, detrimental and scavenging would be an assistance, but it is doubtful if initial temperatures are high enough in actual engines to account for the pre-ignitions which occur, judged in the light of these ignition temperatures measured, and it is, therefore, extremely likely that all heat effects, not necessarily for the entire cylinder, but for some part, are the real causes, and in addition the occasional presence of a certain sensitive proportion between oxygen and either *CO* or hydrogen.

The solution of the problem of controlling pre-ignition resolves itself into three parts: (*a*) Maintenance of proportion of the elements of the mixture to those having the higher temperature of ignition, provided this mixture will still contain enough oxygen to burn all the fuel present; (*b*) care in securing as low an initial temperature of the mixture as possible by maintaining inlet passages cool and purging the cylinder as completely as possible of burnt gases. This also involves the maintenance of early ignition to reduce final release temperatures; (*c*) care in designing the machine so that interior parts shall be as well cooled and as uniformly cooled as possible. A well-cooled cylinder with one spot, such as a nut, poorly cooled may just as well be poorly cooled throughout.

The prevention of explosive waves entirely in engine cylinders seems to be impossible. They can be avoided to a large extent and practically eliminated by giving attention to the form of the combustion chamber and to the method of igniting so as to avoid the generation of successive waves that might superimpose.

The maintenance of uniform cylinder mixture, involving, as it does, first, the correct and positive proportioning of air to gas, and later, the uniform mixing of this primary mixture with the burnt cylinder gases in always constant quantities, is a thing which is absolutely impossible with the present type of engine.

THIRD REPORT TO THE GAS-ENGINE RESEARCH COMMITTEE OF THE INSTITUTION OF MECHANICAL ENGINEERS.

By PROFESSOR BURSTALL.

This interesting report gives the results of a number of tests carried out at Birmingham

with a Premier gas engine used in connection with the power and lighting supply of the building. It was capable of developing 150 b.h.p. at a speed of 170 r.p.m. In order to enable the engine to run at a compression pressure of 200 lb. per sq. in., with charges which were estimated to give an initial pressure of 600 lb. per sq. in., the diameter of the cylinder was reduced to 16 in., and other modifications were adopted, such as the adoption of new air and gas valves, and an entirely new cam arrangement for driving them. Further, as the engine used a scavenging charge a small sampler was fitted to take a sample of the exhaust gas out of the exhaust pipe before the scavenger charge passed into the exhaust.

The engine was so constructed that it could be worked on any one of the three known systems of governing, namely, (*a*) keeping the quality of mixture constant, and varying the amount; (*b*) keeping the quantity of mixture constant, and varying the amount of gas; (*c*) hit-and-miss, or cutting-out of charges. The engine was originally arranged to work on system (*a*), but during the whole of the experiments it was arranged to work on system (*b*). As the tests are all at full power, the difference between the two is extremely small.

The working of the engine is as follows: Starting with the suction stroke, the combined air-and-gas valve is opened to a predetermined point by a pivoted lever under the control of the governor, and a positively driven pecker block, actuated by the half-speed shaft, the governor thus controlling the opening of the air-and-gas valve. The mixture, after passing through this valve, enters through the breech end into an annular casing, which contains the inlet valve, and then into the cylinder itself. After shutting the inlet valve the usual sequence of compression, explosion, expansion, and exhaust follows, but about halfway along the exhaust stroke a second valve, called the scavenger valve, lying alongside the mixing valve, is opened from the lay shaft, and allows a current of cold air from the differential piston to enter into the motor cylinder. This serves the double purpose of clearing out the exhaust products, and at the same time cooling the inner surfaces. During the idle stroke of engine this scavenging charge is simply compressed and expanded in the passages leading up to the mixing and scavenger valves. In order to prevent, as far as possible, any possibility of pre-ignitions occurring through hot surfaces, every part of the engine exposed to the flame is water-jacketed, and in order to estimate the amount of heat rejected through each of these surfaces, the water services are taken from separate measuring tanks, and discharged without a mixture from water from any other surface. The temperatures of discharge were in each case measured by thermometers placed in the outlet pipes. The number of separate services are as follows:—

(1) The barrel. This includes the water round the liner only.

(2) The breech, which includes the water supplied only to the flat end of the cylinder end.

(3) The piston.

(4) The inlet valve, inlet casing, and exhaust valve.

The engine worked entirely with "Mond" producer gas, and high-tension ignition of the "Lodge" type was adopted. The tests were undertaken to determine, in the first place, the thermal efficiencies based on the indicated horse power at various compressions, having regard to the richness of the mixture, and, in the second place, to formulate, if possible, the law connecting efficiency and compression. The heating value of the gas was determined by means of a special arrangement consisting of a modification of the "Junker" calorimeter.

The results of the tests showed that for each compression there was a particular mean pressure which gave the highest economy for that compression. This pressure appeared to range between 85 lb. and 95 lb. for all the compressions, with the tendency to increase as the compression rose. Higher mean pressures than these caused the efficiency to fall off. The rise in efficiency with the compression was very marked in some of the trials, rising from a minimum of 28 per cent. up to a maximum of 39 per cent. After this point the efficiency increased comparatively slowly, reaching the maximum of 43 per cent., and then diminishing to 39 per cent. on the highest compressions of all. This result did no accord with the usual belief that economy increased with compression when a suitable mixture was used. For the particular engine experimented upon, the most economical compression pressure was apparently 175 lb. per sq. in. In order to obtain higher thermal efficiencies by the aid of higher compressions, it would be necessary to increase the stroke of the engine in proportion to its diameter. Very high mean pressures, extending to some 114 lb. per sq. in., were proved to be decidedly uneconomical, the economy falling from 39 per cent. to 32 per cent. in some cases. In the actual tests the maximum pressure that was allowed

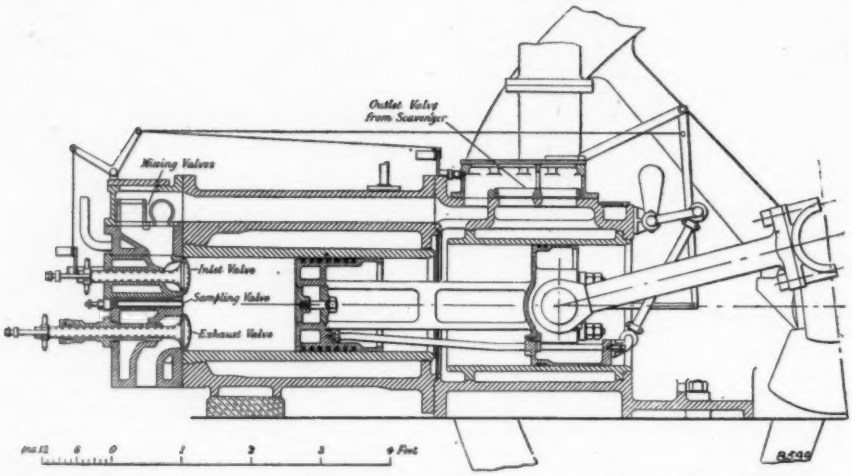


FIG. 1.—16 IN. BY 24 IN. EXPERIMENTAL GAS ENGINE, UNIVERSITY OF BIRMINGHAM. LONGITUDINAL SECTION.

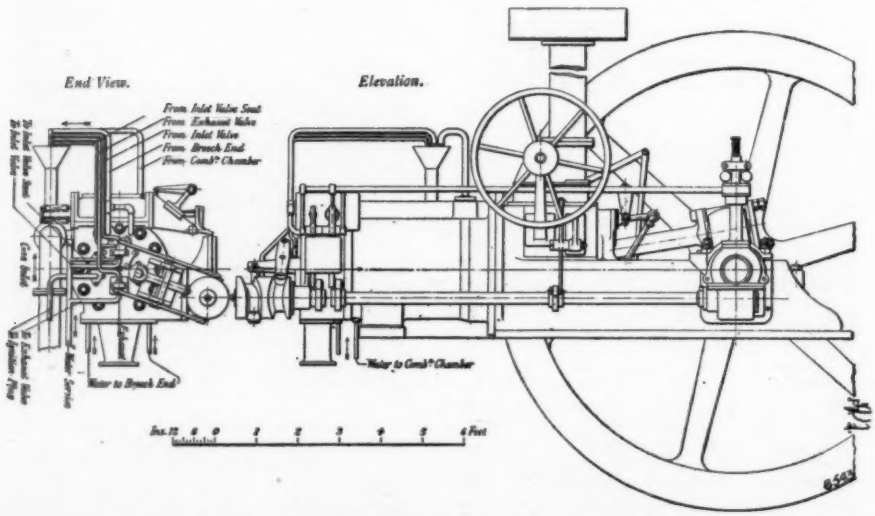


FIG. 2.—16 IN. BY 24 IN. EXPERIMENTAL GAS ENGINE. GENERAL ARRANGEMENT.

was 550 lb., and this was only rarely reached. The whole of the experiments appeared to point conclusively to the fact that the most economical mean pressure was very considerably below the maximum which could be obtained, and that the highest economies were obtained with a comparatively low maximum temperature. That, of course, only applied to the indicated power, and the conclusions as to brake horse power would be widely different. If, however, the engine was constructed to work only with these moderate pressures and temperatures, the whole of the working parts might be very much lightened, and thus a good mechanical efficiency obtained with the very moderate mean pressures.

In reading his paper, Prof. Burstall amplified that part dealing with the point at which further addition to compression reduced the efficiency of the engine. Such a point undoubtedly existed. Confining and compressing a mass of air and gas in the cylinder, there would be gain in the subsequent operations so long as the temperatures produced were not such that more heat was lost from the charge to the walls of the cylinder than was gained during the subsequent operations. In other words, supposing the compression to be carried infinitely high; when expansion was started there would be an infinitely thin layer of gas; all the heat would at once pass into the jacket, and the mean pressure produced would be very small. It was surprising to find the limiting compression came so early as it did. He would not say that the particular economical compression pressure in his engine was applicable to any other; that would depend upon the design of the combustion-chamber. He had no doubt that higher compression pressures would be used with greater economy than he had been able to obtain.

As to how a charge lost its heat to the wall, Prof. Burstall stated that he could not imagine a non-conducting cylinder. All he had assumed was that the charge lost heat according to a law which Holborn and Henning had successfully proved up to 1440° C. He had been forced to assume variable specific heat in the experiments, and had been severely criticised for so doing; but there was evidence to justify him in going somewhat further into that. At all events, at low compressions, according to his computation, the charge lost its heat to the wall very rapidly during expansion, while at the most economical compression the charge lost very little heat to the wall; still further, however, the charge lost a very considerable amount of heat to the wall.

In the ensuing discussion, Prof. B. Hopkinson alluded to the conclusion arrived at in the report, that at a certain point the increase of compression in the engine tested did not lead to any increased efficiency. The experiments at the lower compressions, from 80 lb. up to about 180 lb., added proof to the established conclusion in gas-engine working that efficiency increases with the compression.

Prof. Burstall had, however, inferred from experiments at higher compressions that the efficiency did not increase any further, and that therefore the ratio fell, a result that did not agree with the usual belief. The evidence put forward did not appear to establish Prof. Burstall's conclusions. On comparing the C trials with the A trials, it would be seen that the C trials corresponded to a compression ratio of 7·22. Worked out according to the usual formula, $1 - \left(\frac{1}{r}\right)^{0.4}$, the air cycle efficiency for the compression ratio of 7·22 would be 0·543. In the case of the A trials, with a compression ratio of 8·07, the corresponding value of the efficiency would be 0·566. On the basis of the theory that the gas-engine efficiency went up in proportion to the air cycle efficiency, no great increase in efficiency as between the A and C trials would be expected. The difference was only about 4 per cent. on the amount of work done for a given charge of gas. He felt doubtful, whether, by the methods which Prof. Burstall employed, it would be possible to detect any such increase of efficiency as that. He did not know exactly the degree of accuracy which Prof. Burstall believed himself to have obtained, but would hardly suppose that the author would swear to his indicator diagrams being accurate within 5 per cent., and if that were so, the results obtained might be interpreted as showing that in this engine the efficiency was substantially the same for the C and the A trials. From the evidence as it stood, it was not possible to infer that there was really any departure from the law—it might be so described—that efficiency went up with compression. The evidence certainly was not conclusive.

Prof. Burstall had given as one of the reasons for adopting the scavenger engine a desire to avoid pre-ignition, and had thought that he would avoid it by the cooling action of the scavenging charge on the cylinder walls. He questioned, however, whether, in the case of a completely water-jacketed engine, the inner surface of the

walls would rise, with or without scavenging, to a temperature in excess of 200° C.

It was different in the case of an uncooled piston; in that case the heat given to the piston had to travel a long way, and the piston might get very hot; but, even so, the speaker much doubted whether scavenging would have very much effect in cooling the inner surface. He had been lately measuring the actual temperature of the inner surface of the uncooled piston in a fairly large engine by putting a bolt through the piston, and having a nickel-iron thermo-couple on the inner surface, the couple thus registering the temperature of the piston. With this arrangement in a hit-and-miss engine it was possible to get an idea of the temperature effect of scavenging, and it did not appear to be of great practical importance.

Other speakers referred to the means Prof. Burstall had employed of determining the point of ignition, the time given for the ignition appearing to be much too early.

In his reply, Prof. Burstall said that one or two misconceptions had arisen, the question of the firing point being one. On his original drawing the speaker had marked it "Time passage of the primary spark," but the committee, considering the wording lengthy, had altered it to "ignition-point." Personally, he did not know where the ignition-point was; he had merely indicated the time when the primary spark at the lay-shaft was completed; no doubt the actual spark was later.

With regard to Prof. Hopkinson's statement, he agreed with his conclusion so long as the air efficiency was confined to the air-engine, but not when applied to a gas-engine. The expression Prof. Hopkinson has employed could only be proved on the basis of constant specific heat; it could not be proved when the specific heat varied, as in Holborn and Henning's results, where the specific heat was a quadratic function of the temperature. The speaker could not see that the accuracy or otherwise of the indicator made any difference. Lower results were persistently obtained in the case of the A and B trials than in the C trials. The engine had been tested very often by students and others, with all sorts of indicators, and uniformly the B trials were inferior. If there were a permanent error of 5 per cent. in the indicator, it was not easy to understand why it should come out on one test and not on the others. He could not agree with Prof. Hopkinson that the temperature of the surfaces exposed to the hot gases rarely exceeded 200° C. He had taken pieces of carbon ¼ in. thick out of the cylinder—absolutely hard carbon, which could never have been formed at 200° C. That trouble he had not alluded to in the report. In his mind there was no doubt that these pieces of carbon must have been exposed to 500° or 600° C. He could produce premature ignitions of the engine in a moment by merely shutting off the water supply, even from one of the valves. That alone indicated the high temperature of the inner surface of the walls.

New Road Bridges over the Nile at Cairo.

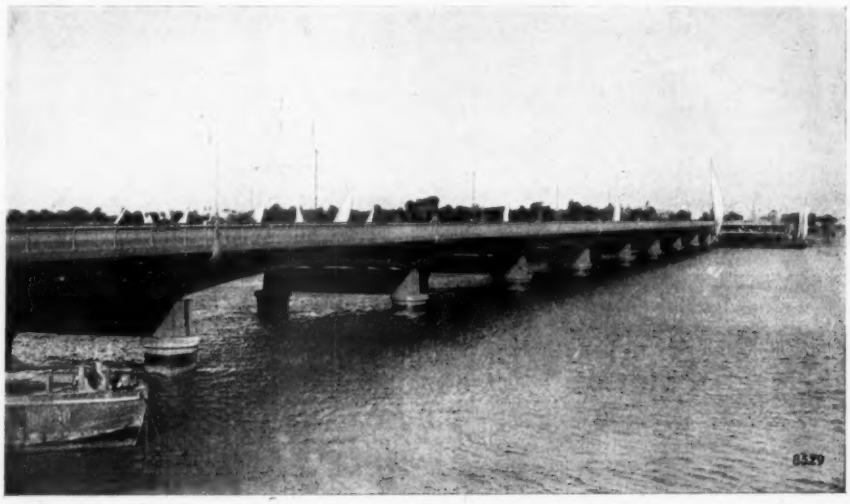
In the summer of 1903, at a time when so much was heard about the inability of British manufacturers to compete with their continental rivals, specifications were issued by the Egyptian Government inviting tenders and designs for three road bridges to be constructed over the Nile at Cairo. The limit of time by which tenders and designs were to be deposited with the Egyptian Government in Cairo was February 1st, 1904. No less than thirty-eight designs for each bridge were submitted, and tenders were received from thirteen firms, six of whom were French, one each German, Belgian, Italian and Swiss, and three British, one of whom submitted an American design. It will be seen that the competition was international, and keenly competed for, as the Belgian firm alone submitted nine different designs for the large bridge and three for each of the small bridges. As many as sixty to seventy drawings accompanied each tender, and in many cases general and detailed perspective drawings were attached to them. It was estimated that the preparation of the designs alone represented an expenditure of about £25,000.

A Commission was appointed to examine the designs and calculations and report to the Government. After a careful and prolonged examination of the various proposals a gratifying result was achieved by the acceptance of the designs and tender of the joint firms of Sir William Arrol & Co., Ltd., of Glasgow, and Messrs. Head, Wrightson & Co., Ltd., of Middlesbrough, for constructing the three bridges at a cost of £192,000. The additional work in connection with the roadways and abutments

ordered after the contract was signed amounted to about £30,000.

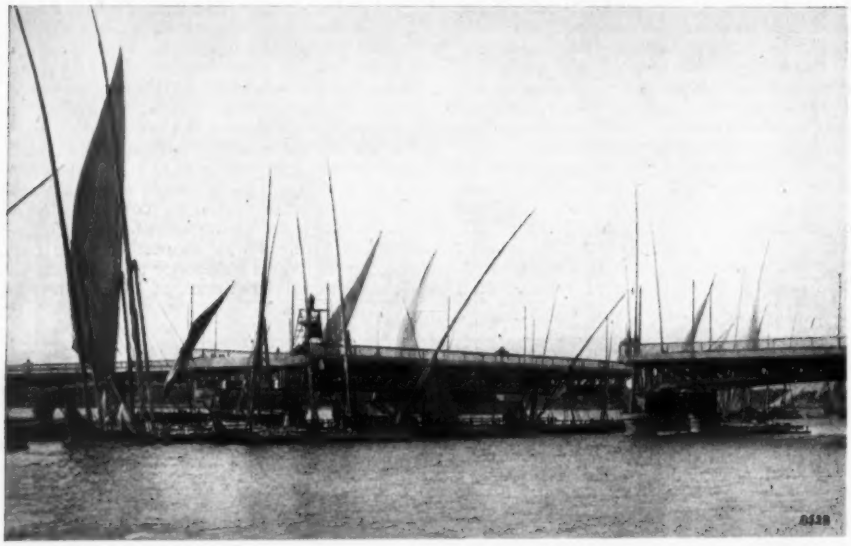
The main bridge, which is the largest and finest bridge on the African continent, crosses from Ghizeh near the road to the Pyramids to the Island of Rodah, and the two smaller bridges from Rodah Island to the main road leading to old Cairo.

The successful designs were prepared by the Civil Engineering Staff of Sir William Arrol & Co., Ltd., under the able direction of the late Mr. J. E. Tuit, M.Inst.C.E., whose untimely death before the completion of the work is much to be regretted. The designs were completed by Mr. Adam Hunter, A.M.Inst.C.E., to whom we are indebted for the following information, and who had been associated with Mr. Tuit in the early part of the work.



VIEW SHOWING THE 535 METRE BRIDGE.

In designing the bridges special attention was given to the requirements of the specification stating that "by reason of the position of the bridge, the æsthetic effect of the design is of material importance." It was considered that many people would regard the larger bridge not merely as a means of crossing the river, but as offering special facilities to enjoy the scenery and cooler atmosphere in the vicinity of the Nile, and consequently it was most desirable that no portion of the structure should in any way interrupt a clear view being obtained of the river and city of Cairo. A design was therefore chosen which admitted of the whole of the superstructure being placed below the roadway, and by curving the underside of the main girders between the piers, the structure, when viewed from the river banks, would have a graceful appearance. Although the design of the structure was kept as simple as possible, and a pleasing appearance obtained by making the structure correct from a scientific point of view, suitable architectural treatment was adopted for the cast iron parapets and masonry abutments. A conventional treatment of the palm was introduced into the design of the parapets and the long straight line of the handrail cope broken up at the piers by suitably designed lamp standards. The masonry abutments at the entrances to the main bridge have substantial masonry pilasters of an Egyptian character designed to be in accordance with the dignity of the undertaking. Provision is made that suitable statuary may be placed on the pilasters if considered desirable by the Egyptian Government. The principle adopted throughout the design was to adhere to a form which was structurally correct and that any architectural treatment should be such as would emphasise the construction and give suitable expression to the structure. This



OPENING SPAN OF LARGE BRIDGE.

bridge might be cited as an example of the successful co-operation of the engineer and the architect. The general effect of the structure is pleasing, and when viewed from the river or its banks, it has a graceful appearance. The masonry abutments and cast iron parapets were designed by Mr. Roland W. Paul, of London, who advised generally on the æsthetic treatment of the structure.

The main bridge is 1,755 ft. in length between abutments divided into ten spans of 140 ft., two end spans of 70 ft., and a double swing span 220 ft. in length with two clear openings of 66 ft. The bridge is 20 metres in width, or 66 ft., between parapets, divided into two footpaths 8 ft. wide, and roadway of 50 ft. in width, of which 16½ ft. is allocated to the tramway zone and 32 ft. 6 in. to the road zone.

There are five main longitudinal girders of lattice construction in the cross section of the bridge, with cross girders between them and longitudinal floor girders to carry the roadway, and projecting cantilever brackets on the outer main girders to carry the footways. The longitudinal main girders are continuous over the piers, but are cut in alternate spans at the two points of contra flexure, which arrangement not only facilitates the correct computation of the strains in the structure and the provision for movements due to temperature and settlements of the pier foundations, but also adds considerably to the stiffness of the girders which were made of shallow depth in order to escape the reach of floods.

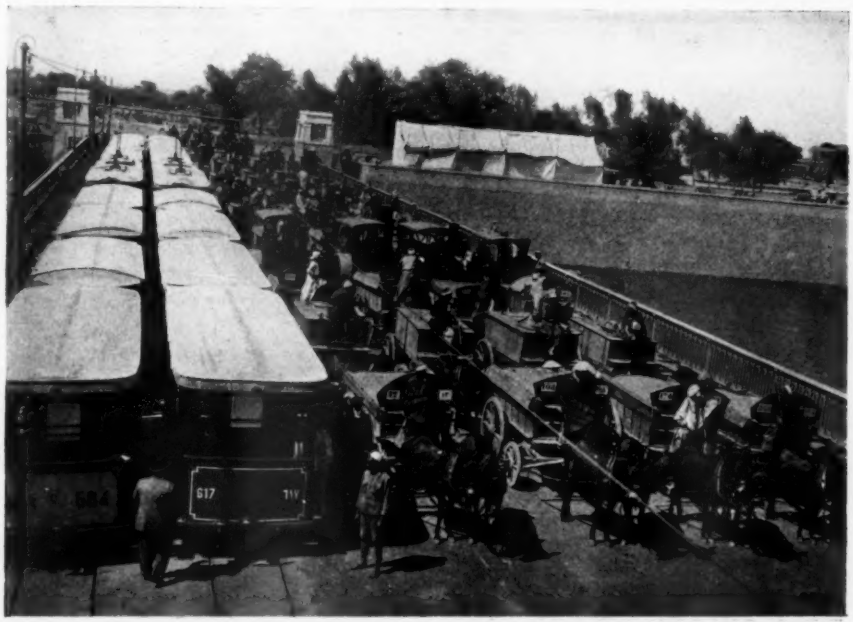
Each main pier is formed of pairs of steel cylinders 18 ft. diameter at the base, reducing to 10 ft. 6 in. at the top, and placed 47 ft. 3 in. apart centres. The cylinders in all cases are 96 ft. in length, the bottom being 72 ft. below Low Nile. They are connected together at the top by means of a heavy capsill girder 7 ft. 6 in. deep and 4 ft. wide, which serves to carry the longitudinal main girders of the bridge and to bind the two cylinders rigidly together to resist the lateral forces from wind and other causes. The lower portion of each cylinder to within about 12 ft. below the river bed is formed of steel plates, and the remaining portion of cast iron, 1¼ in. in thickness. It was considered that the exposed portion, being subject to the greatest corroding influences due to being alternately wet and dry, would become unsightly, and that cast iron should be adopted on account of its quality of resisting corrosion better than steel. The interior of the cylinders is filled with cement concrete of an excellent quality, composed of four parts

of Abu-Zaabel Basalt, two parts of sand, and one of portland cement. The concrete hearting alone was proportioned to carry the whole of the loads without allowing for any assistance from the cast iron or steel forming the cylinder.

In drawing up the programme of operations for each season at the site, it was intended to begin sinking operations with the piers at the beginning of December, 1904, and complete the piers by the end of June, 1905. . Through delays at Alexandria the steelwork for the cylinders did not reach Cairo until January 7th and 20th, the arrival in Alexandria having taken place on December 1st and 2nd respectively. Through these delays the sinking was not commenced until two months later than had been expected.

The aggregate amount of sinking for the main piers was 1,600 lineal ft., equivalent to about 16,000 cub. yds., and this was done under air pressure. The excavation was principally in grey sand. The sinking was started in the beginning of March, 1905, and completed in May, 1906, the work having been carried on continuously with the exception of the months of September and October, when no sinking was done on account of the High Nile flood. The foundation work was therefore completed one month ahead of the time notwithstanding the early delays in starting the work. The total time occupied in sinking the piers was therefore about fifty-six weeks, giving an average rate per week of 28½ lineal ft. or 285 cub. yds. The whole of the sinking was done by native labour under European supervision. The Arab labour on the whole was satisfactory for the work of sinking the piers, and they stood the air pressure very well, which reached as high as 35 lb. per sq. in. above the atmosphere. The lower parts of the cylinders were built and riveted in the yard adjoining the bridge, and were taken along the temporary stage and lowered by means of hydraulic jacks to the bed of the river. Previous to lowering to the bed of the river the cylinders were lined with concrete, after which the air shafts and locks were fixed in position.

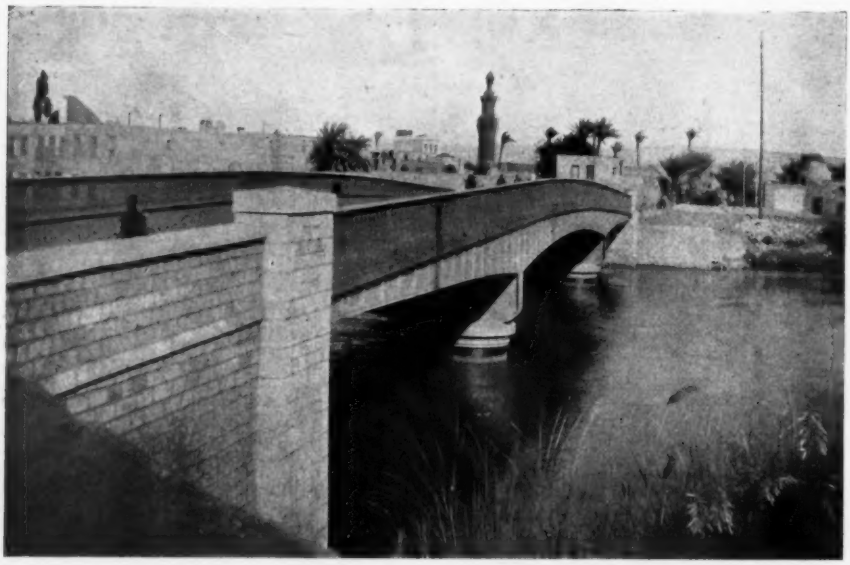
The main girders were built *in situ* on the temporary stage, which had been carried out from the shore as the sinking of the



SHOWING TEST LOADS UPON LARGE BRIDGE.

cylinders was proceeded with. Considerable difficulty was experienced in getting skilled labour for the work of building and riveting the steel superstructure. The general prosperity of the country caused suitable native labour to be scarce, and such as could be obtained had to be specially trained under European foremen to work the pneumatic and hydraulic riveting machines. Over half-a-million rivets had to be put in at the site, and great credit is due to the staff for the careful supervision exercised, that the total percentage of defective rivets which had to be replaced did not exceed 1 per cent. for hydraulic, 2 per cent. for pneumatic, and under 1 per cent. for hand riveting. This percentage is almost the same as obtained at home with skilled white labour. With regard to this point, the late Sir Benjamin Baker reported to the Egyptian Government that the work done by the native riveters was of a satisfactory quality, and that, "owing to the demand for riveters on other works in Egypt, the contractors had not only to train their own men, but also practically to train men for other works where contractors did not hesitate to pay whatever sum might be necessary to secure trained riveters from the Rodah Bridge. This double training of men was no doubt somewhat of a hardship to Messrs. Sir William Arrol & Co., Ltd., but it was an advantage to the country, having regard to the demand for riveters." In Lord Cromer's last report to Parliament on Egypt and the Soudan, he states that Sir William Garstin writes, "The steelwork has come together very well, and the quality is excellent."

The swing span consists of five main longitudinals as in the ordinary spans: having an overall length of about 220 ft. and weighing complete about 1,000 tons. When open it allows two clear passages for vessels of 66 ft. The main girders rest directly upon a circular drum girder 47¼ ft. diameter and 19 in. deep carrying the upper cast steel roller path. The lower cast steel roller path is supported by the four main capsill girders of the pivot pier, which form a square with short diagonal girders at the corners. The span turns upon a ring of 105 live rollers of forged steel 14 in. mean diameter and 13½ in. in length, which are secured to the peripheral girders. Rigid radial arms attach these girders to the castings round the centre pin. The span is operated by two 30 b.h.p. electric motors acting through gearing on to the



VIEW OF THE 67 METRE BRIDGE.

VIEW OF THE 83 METRE BRIDGE.

rack, which is $3\frac{3}{4}$ in. pitch and 8 in. broad, and securely bolted to panels cast on the lower roller path. One 15 b.h.p. electric motor is placed at each end of the swing span to operate the toggling gear for raising the ends. The bridge can be opened or closed in three minutes, including the time for all operations necessary in connection with the swinging. In addition to the complete electrical equipment for swinging the bridge, separate gearing is provided that the bridge may be opened or shut by manual power in case this should be sometimes necessary.

The masonry abutments and pilasters are built of lime-stone from the Bassatine Quarries, near Cairo. The stone was rough dressed at the quarries before being brought to the site. The abutments are built of rubble masonry with rock-faced facework and fined axed quoins, the whole being set in cement mortar of the proportions of four parts of sand to one of cement. Dressed ashlar was used in the string course and for the upper part of the pilasters. Each abutment is founded upon six steel cylinders filled with cement concrete. The bottom of these cylinders is 69 ft. below High Nile in the case of the Rodah abutment, and 56 ft. in the case of the Ghizeh abutment. The original proposal was to found the abutments upon timber piles, but from the experience gained in making the temporary stage this was found to be impracticable.

Proposals were submitted to the Government by the contractors to substitute steel cylinder foundations in place of piles, and this proposal was carried out.

The roadway over the bridge is formed of concrete supported on buckled plates and covered with 2 in. of asphalte. The footpaths were specified to be Karamania pine, but this was changed towards the end of last year to 1 in. of asphalte upon flat steel plates.

The two small bridges are of plate construction and in three spans, the abutment spans being about three-quarters of the centre span. There are seven main girders in the cross section of each bridge and are all placed below the roadway. The main girders are continuous with the two points of contrary flexure cut in the centre span. The two centre piers are each formed of a pair of cylinders 12 ft. in diameter at the base and united together at the top with a substantial capsill girder. The masonry abutments are founded upon brick walls which were sunk by native divers. The roadway and footpaths are similar in construction to the main bridge.

The erection of the steel superstructure was started in January, 1906, as originally proposed, and was completed by the end of December of the same year, fully two months ahead of the forecasted time of the programme made out at the beginning of the work.

A double line of tramway of one metre

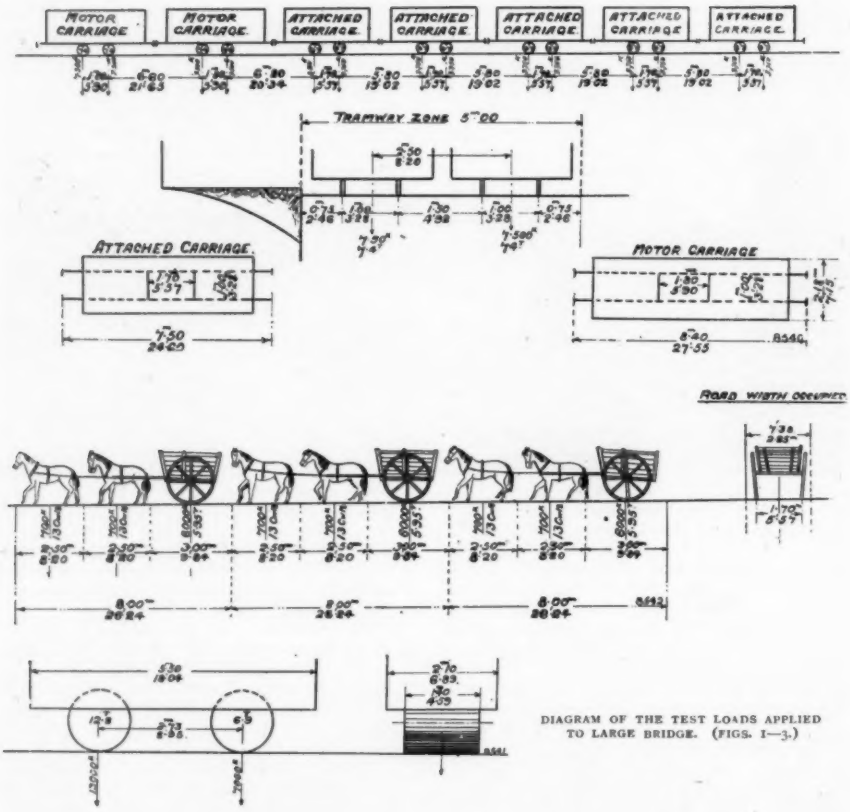


DIAGRAM OF THE TEST LOADS APPLIED TO LARGE BRIDGE. (FIGS. 1—3.)

gauge passes over all the three bridges and connects up the Cairo lines with the one to the Pyramids. The bridge has been officially tested by the Government, with most satisfactory results.

The footpaths for the whole length of the large bridge were covered with 1,000 tons of sand, and while this dead load was on the bridge a moving load of 450 tons, made up of loaded carts, tramcars, and traction engines, was passed over from end to end several times. This heavy moving load was rested on each span while records of the deflections were taken. The structure passed successfully under this severe test load, which was such as could never be brought on the bridge again unless it was specially arranged, as it took days to collect the vehicles. The deflections of the girders were very much under the estimated amounts, indicating material of an excellent quality and riveting and other workmanship of the highest class.

Specification.—The whole of the steelwork was manufactured by the Siemens' Martin open hearth acid process. The structural steel has an ultimate tensile strength of not less than 28 tons per sq. in., and not more than 32 tons with an elongation of at least 20 per cent. in a length of 8 ins.

In addition to the permanent weight of the structure, the following loads were taken in designing the bridge:—

(*a*) A distributed dead load of 400 kilogrammes per sq. metre (82 lb. per sq. ft.) covering the whole surface of the roadway, including the tramway zone and the two footways.

(*b*) Two lines of tramcars of unlimited length moving in the same or opposite directions (Fig. 1).

(*c*) A traction engine of 20 tons weight (Fig. 2).

(*d*) Three lines of loaded carts of unlimited length occupying the roadway zone while the tramways are on the bridge and the footpaths loaded. All loads were moved simultaneously along the bridge (Fig. 3).

(*e*) A wind pressure of 150 kilogrammes per sq. metre (30·72 lb. per sq. ft.) when the bridge is loaded with the full dead and live loads.

(*f*) A wind pressure of 250 kilogrammes per sq. metre (51·2 lb. per sq. ft.) when the bridge is unloaded.

The surface exposed to the wind pressure was assumed to be 50 per cent. in excess of that seen in elevation to allow for the pressure exerted on the leeward girders, and the surface of the moving load was taken at 8 sq. ft. per foot run of the bridge.

The following unit working stresses were adopted under these loadings :—

Main Girders and Capsill Girders.

In tension, flanges : 8·5 kilos. per sq. millimetre of net section = 5·4 tons per sq. in.

In tension, web ties : 6·0 to 8·5 kilos. per sq. millimetre of net section = 3·8 to 5·4 tons per sq. in.

In compression, flanges and struts : 80 per cent. of the permissible tensile stresses in the corresponding members calculated on the gross section.

Alternate stresses for tension : 6·0 to 8·5 kilos. per sq. millimetre of net section = 3·81 to 5·4 tons per sq. in., according to the formula, $8{\cdot}5\left(\frac{\text{min. stress}}{3\frac{1}{3}\text{ max. stress}}\right)$.

Alternate stresses for compression : 80 per cent. of the corresponding permissible tensile stress.

Rivets, shearing : 6·8 kilos. per sq. millimetre = 4·32 tons per sq. in.

Rivets, bearing : 16 kilos. per sq. millimetre = 10·16 tons per sq. in.

Hob plates, shearing : 5 kilos. per sq. millimetre of gross section = 3·18 tons per sq. in.

Cross Girders and Stringers.

In tension, flanges or in extreme fibres in bending : 7·5 kilos. per sq. millimetre of the net section = 4·76 tons per sq. in.

In compression : Same gross area as the tension flanges.

Shearing in web plates : 4·5 kilos. per sq. millimetre of the gross section = 2·86 tons per sq. in.

Rivets, shearing : 5 kilos. per sq. millimetre = 3·18 tons per sq. in.

Rivets, bearing : 12·75 kilos. per sq. millimetre = 8·1 tons per sq. in.

Bearings at Ends of Suspended Girders.

Expansion link, in tension : 7·25 kilos. per sq. millimetre of net section = 4·6 tons per sq. in.

Bearing pressure on moving surface of pin for expansion link : Same as tensile stress per sq. in.

Bearing pressure on fixed bearing of pin for expansion link : 1½ times tensile stress per sq. in.

Shearing stress in pin : Three-quarters of tensile stress per sq. in.

Bending stress in pin : 1½ times tensile stress per sq. in.

Rocker at fixed end (cast steel) bearing pressure : 5 tons per lineal inch.

Bearings on Abutments.

Pressure on masonry bedstone : 10 kilos. per sq. centimetre = 9¼ tons per sq. ft.

Tension in anchor bolts : 5 tons per sq. in. at bottom of thread. The length of bolt must be sufficient to lift 1½ times the maximum upward forces.

Foundations.

Compression on cement concrete in cylinders and on masonry in abutments : 10 kilos. per sq. centimetre = 9¼ tons per sq. ft.

On grey sand under cylinders : 6 kilos. per sq. centimetre = 5½ tons per sq. ft. Nothing is allowed for buoyancy or material displaced.

On 12 in. × 12 in. pitch pine piles : 30,500 kilogrammes = 30 tons per pile.

Timber joists (Karamania pine) : 60 kilos. per sq. centimetre = 7·62 cwts. per sq. in.

The various operations in connection with the opening span were carried out within four fifths of the specified time. The construction of this bridge is most creditable to the joint contractors, Messrs. Sir William Arrol & Co., Ltd., and Head, Wrightson & Co., Ltd., and in support of the statement it is only necessary to quote from the report of the late Sir Benjamin Baker, in which he states "I have to-day, February 1st, 1907, inspected the above bridge and am able to certify that as regards material and workmanship it is entirely satisfactory, and that in all respects the structure is a fine example of the high class of work which can now be turned out by bridge builders of great experience, whose manufactories are supplied with all the most modern appliances." The chief direction of the work both at home and in Egypt was under the able control of Mr. Andrew S. Biggart, and the joint firms were represented at the site by Mr. William N. Bakewell, M.Inst. C.E., and his assistant Mr. W. Burnside. The work in Egypt was carried out under supervision of Mr. Arnold Perry, Under Secretary of State for Public Works, whose Resident Engineer was Mr. A. T. Knowles.

Torsion-Meters, as applied to the Measurement of the horse power of Marine Steam Turbines.

*By J. HAMILTON GIBSON.**

WHEN a revolving shaft transmits power it always twists slightly throughout its length. In other words, the end at which the power is applied moves slightly in advance of the end where the work is done, the amount of twist varying directly as its length, directly as the moment of the load applied, inversely as the rigidity of the material, and inversely as the fourth power of its diameter, the formula reading :—

$$\theta = \frac{10{\cdot}2TL}{GD^4}$$

where θ is the angular displacement in radians, T = twisting moment in inch pounds, L = length of shaft in inches, G = the modulus of rigidity, and D = diameter of shaft in inches. The law holds good absolutely for all shafts which are not stressed beyond the elastic limit. As shafts are usually designed with a large factor of safety, it follows that the amount of twist, or the "torque," as we prefer to call it, is very small. In propeller shafting, for instance, the torque is really more than 1 deg. for 10 ft. of length, so that for a 12 in. shaft the circumferential displacement is only about $\frac{1}{8}$ in. at full power.

Various methods and numerous instruments have been devised to enable an observer to read off the torque of revolving shafting, and such instruments are rightly termed "torsion-meters," or, if self-registering, "torsion-indicators." Many of these instruments are extremely ingenious, and it is proposed in this paper to briefly examine and describe some of them.

The rapidly growing adoption of steam turbines for ship propulsion has created a demand for some ready means of ascertaining their horse power, and as the steam engine indicator is not suitable for this purpose, we are thrown back on a torsion-meter as the only known method by which such information can be obtained. The power of a steam turbine may be estimated approximately by calculating the amount of water passed by the feed pumps, or by measuring the number of heat units that pass through the turbines in a given time; but a co-efficient of efficiency must be first determined, and no account is taken of the revolutions in such estimates. As, however, "revolutions" is the very essence of power in dealing with the question of ship propulsion, that would be a very unsatisfactory method of reporting the power from a shipowner's point of view. How often do we hear the claim made that so-and-so's feed-heater, for instance, has given a liner an extra one or two revolutions on the same coal consumption as before? Observe there is no mention of power, because it is recognised that under similar conditions, the maintenance or improvement of the revolutions is the only thing that matters.

Now it is a well-known fact that a turbine, unlike a reciprocating engine, passes almost as much steam when standing as when revolving at full speed; and it has therefore become an almost imperative necessity, in fixing the responsibility as between the boiler and the turbine, to know what power the turbine is transmitting to the propeller under varying conditions. The power thus ascertained is called the "shaft horse power," in contra-distinction to the term "indicated horse power" which has come to be applied exclusively to the results obtained by "indicating" the mean pressures in the cylinders of a reciprocating engine. In this connection, "brake horse power" and "shaft horse power" are, of course, identical.

A small propeller working deeply immersed in smooth water is a fairly uniform brake, and the turning moment of a steam turbine is also very uniform. Consequently there is little, if any, fluctuation of the torsional stresses in the propeller shafting. If, then, we can ascertain the torque at only one point in each revolution, it may be assumed that, knowing the revolutions, we have all the information required to calculate the work done. It is very

* "Proc. N.E. Coast Inst. of Engineers and Shipbuilders."

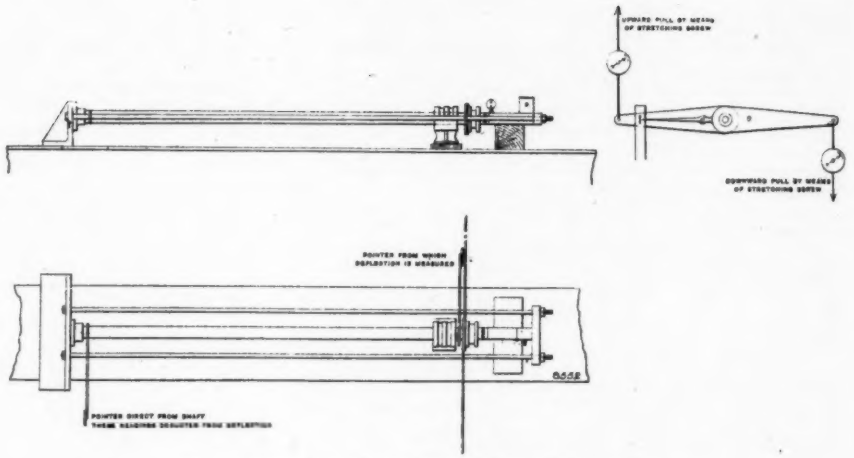


FIG. 1.—ARRANGEMENT FOR TESTING TORQUE OF SHAFTING IN SHOP.

different, however, in the case of reciprocating engines. The turning moment is anything but uniform, there are several points of maximum and minimum torque in each revolution: in fact, it is not an unknown experience to find that at one or more points in each revolution the torque is negative, that is, the propeller, acting as a fly-wheel, overruns the engine and actually pulls the engine round after it. In all cases of reciprocating engines, therefore, it becomes necessary to read off the torque at several points in the revolution, the more points the better. The mean torque is then taken in making calculations of power. For a clear appreciation of the problem of torque measurement it is expedient to keep the foregoing facts well in mind, and principally to remember that we are dealing with extremely minute angles: for it is no exaggeration to say that an error of a hair's breadth may mean a difference of several hundred horse-power in the result.

Before applying any form of torsion-meter to a shaft, we must know its "modulus of rigidity," that is, how much it will twist with a given static load applied at the end of a lever of known length. This can only be done satisfactorily in the workshop, preferably on a long rigid lathe bed (Fig. 1). One end of the shaft is securely fixed and a twisting moment applied at the other end. To eliminate the effect of friction in the supporting bearing at the free end it is advisable to use two levers, one at either side, as shown in the illustration, and the loads are preferably applied by graduated spring balances. Two pointers independent of the load levers are secured to the shaft as far apart as practicable, and the difference in the angular movement of these two pointers gives the true twist for that length of shaft. If the pointers are made 57·3 in. long from the shaft axis, their ends will describe 1 in. of arc for 1 deg. of twist, and a decimally-divided straight edge will then measure the twist to within $\frac{1}{100}$ of a degree, which is quite near enough for all practical purposes, and we can proceed to calculate the modulus of rigidity from the formula.

Observe that a propeller shaft is subject to two distinct stresses. Not only is it twisted as between the engine and the propeller, it is also compressed longitudinally by the propeller thrust, the compressive stress being sometimes as much as 20 per cent. of the shear stress at the surface of the shaft, produced by torsion alone. This compression augments the torque by an appreciable amount, which has been actually measured in numerous experiments, and may be taken roughly as 3 per cent. for hollow shafts and 1 per cent. for shafts which are solid. It might be considered sufficient to calibrate only one shaft in a multiple screw vessel; but it is found that similar shafts, with identical tensile and elongation tests, have different moduli of rigidity, probably due to their varying elastic limits and some slight difference of homogeneity in the material. The only way therefore to ensure accuracy is to calibrate each shaft separately

and to build up a power diagram, as shown in Fig. 1, for each.

Another point to bear in mind is that a working propeller shaft is "alive," and this condition must be imitated as far as possible during calibration by jarring the shaft with repeated blows of a mallet so as to keep the mass in a state of molecular vibration. Otherwise the phenomenon of mechanical hysteris, so marked in some static experiments, will obtrude itself and vitiate the results.

Having established the true modulus of rigidity for each shaft, we may proceed to build up our power diagrams based on the formula,

$$H = \frac{\theta D^4 N}{CL},$$

where H = shaft horse power, θ = torque in degrees, D = diameter of shaft in inches, N = number of revolutions per minute, C = constant varying with the modulus of rigidity, and L = length of shaft in inches. In this formula we have all the elements for obtaining the shaft horse power, and it only remains to ascertain the number of degrees of torque by means of a reliable and accurate torsion-meter. Naturally a mechanical engineer would employ mechanical means for the purpose, in the first instance at any rate, and we will describe two such means that have been tried with varying success.

Dr. Fottinger's apparatus (Fig. 2) has been used on several German boats, and consists essentially of two stiff tubes encircling but free of the shaft, except at their remote ends, where they are rigidly secured to the shaft. The free ends of each tube are brought together and terminate in a pair of disks, the disks revolving with the shaft in parallel planes. Assuming the disks to be 2 ft. diameter and the two points on the shaft to which the tubes are secured to be 10 ft. apart, the edges of the disks will then have about $\frac{1}{4}$ in. movement relative to one another at full power. Means are introduced to multiply this movement by the employment of links and levers, and the torque is recorded by an indicator pencil moving round a fixed paper cylinder concentric with the shaft. When there is no torque the line drawn by the pencil is a continuous circle in the same plane, and this line represents the zero or base line from which the subsequent torque indications are measured. When the shaft transmits power "ahead," the indication for varying torque, as in a reciprocating engine, is a wavy line on one side of the base line. For "astern" power the indication is, of course, on the opposite side. The back lash of the link work is taken up by light springs to steady the pointer or pencil.

Several diagrams taken by a "Fottinger" meter are shown in Fig. 3, which exhibit clearly the fluctuating torque of shafting driven by recriprocating engines, and at full power a negative torque is seen, as previously referred to, at that period of the revolution where the propeller overruns the engine.

Another form of mechanical torsion-meter is that of Mr. Collie's (Fig. 4). Instead of tubes encircling the shaft, which are limited in length by the distance between couplings and the plummer block bearings, two light counter-shafts parallel to the main shaft and driven from it at their remote ends by sprocket wheels and chain gearing are carried overhead. Their free ends are screwed into each other, one of them forming

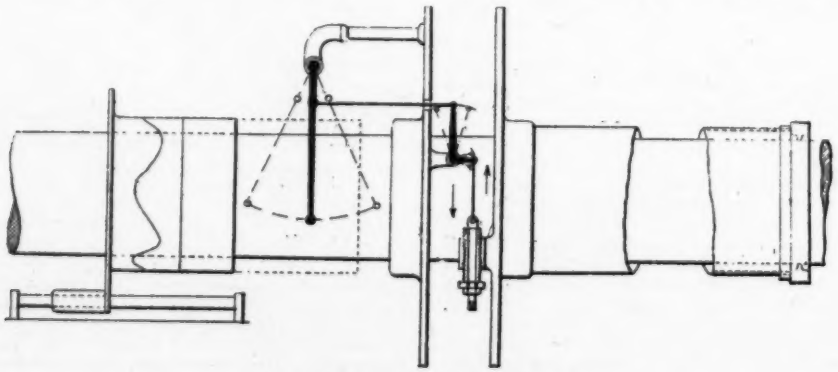


FIG. 2.—THE "FOTTINGER" TORSION-METER.

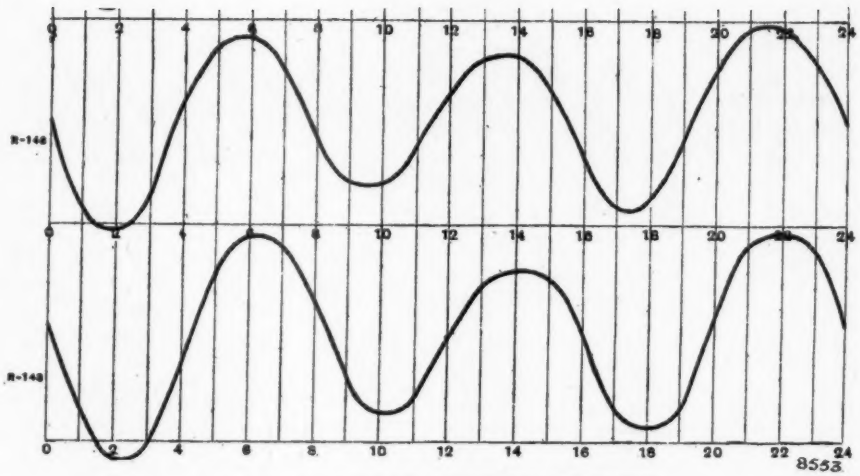


FIG. 3.—DIAGRAMS TAKEN WITH A "FOTTINGER" TORSION-METER.

the nut and having a limited longitudinal movement, whilst the other has none, merely revolving in a small thrust block. As one end of the main shaft revolves slightly in advance of the other, the countershafts screw themselves into or out of one another according to the direction of rotation of the main shaft by an amount depending on the power transmitted. The longitudinal movement is transferred to a pointer and rendered visible on a dial which is graduated on either side of the zero into so many degrees of torque "ahead" and "astern." The back lash of the gear is taken up by springs as in the "Fottinger" meter, and it is a simple matter to add a continuous recording apparatus if such be required.

As a variant on the concentric tube and countershaft methods of measuring the twist of a shaft by means of a parallel member not exposed to torque, several inventors have made use of the fact that some main shafts are hollow, and fit an inner shaft loosely fitting the bore (Fig. 5). One end of the inner shaft is secured to the main shaft, and to the other end is fitted a pointer or spider, the radial arms of which emerge through grooves cut in the face of a coupling between the coupling bolts. The spider shows the same movement as the remote fast end of the inner shaft, and moves relatively to the coupling at which it emerges. Various devices are adopted to show and record the relative movement which, of course, gives the torque of the main shaft for the length of the inner shaft.

The best known electrical torsion-meter is the "Denny-Johnson" apparatus (Fig. 6), which has been frequently described in technical publications. Briefly, it is made up of two revolving armatures secured to the shaft as far apart as possible. Each

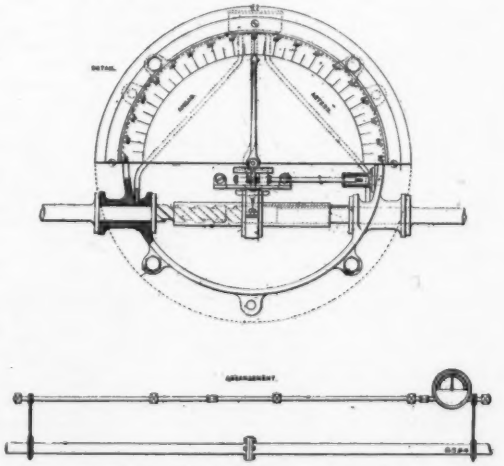


FIG. 4.—COLLIE'S MECHANICAL TORSION-METER.

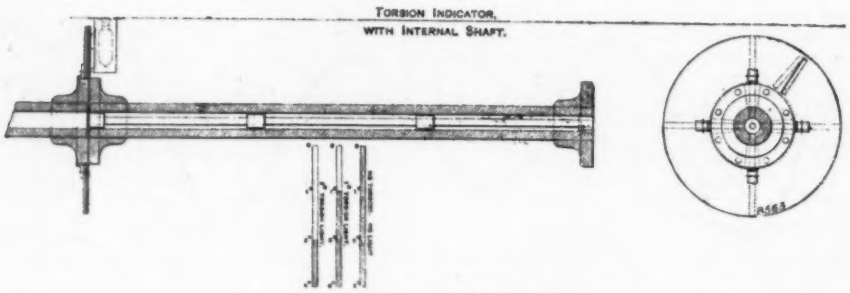


FIG. 5.—TORSION INDICATOR WITH INTERNAL SHAFTS.

armature had a pointed or chisel-shaped magnet which moves over but does not quite touch a finely wound coil. The coils are connected up in a series through a "Wheatstone" bridge arrangement to a telephone receiver. When no power is being transmitted the relative positions of the revolving magnets and coils are identical at each revolution, and no sound is heard in the telephone receiver. But when the shaft twists, the armatures get "out of step," as it were, and a clicking sound is heard until the pointer in the recording box is moved by an amount equal to the number of windings in the coils, indicating that one magnet is ahead of or astern of the other, until silence again ensues, and thus the angle of torque is caught and measured.

(To be continued.)

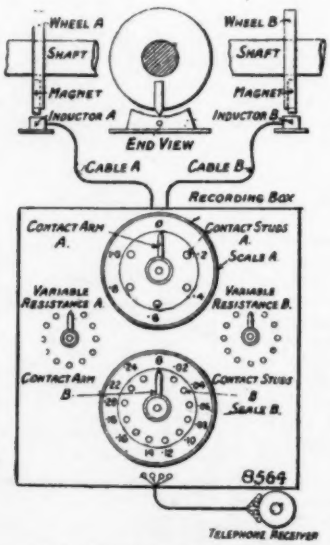


FIG. 6.—"DENNY JOHNSON" APPARATUS.

RAILWAY NOTES

The Work of the 4-6-0 Type Compounds on the Eastern of France Railway.

The Eastern of France Company put into service at the commencement of 1896 the first passenger locomotives of a new type for special duties in connection with the international trains which are frequently exceptionally heavy, and, moreover, have very fast timings. The new locomotives, of which thirty-two are actually in service or completed, are replacing the 4-4-0 type which were found inadequate for hauling trains exceeding 300 tons (m) in weight at the higher speeds required. A further batch of eighteen of the 4-6-0 type are now under construction, and will complete the series of fifty, all of which have been constructed at the works of the company at Epernay. An example of the class, it may be remembered, was shown at the recent Milan exhibition. The batch, numbered from 3103 to 3132, are employed on the express services of the three main lines: Paris to Belfort, Paris to Nancy, and Paris to Charleville, and have enabled the times between these places to be sensibly reduced, with increased train weights.

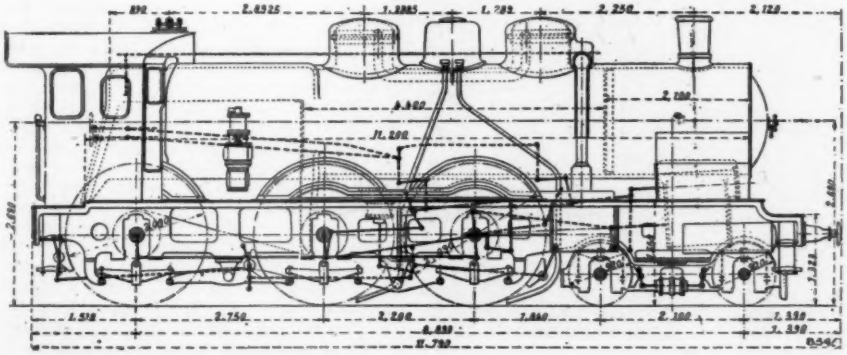
From Paris to Belfort (443 km. = 277 miles) a short time back occupied 6 hrs. 10 min. with a load of from 240 to 250 tons; to-day the timing has been reduced to 5 hrs. 22 min. with loads of from 280 to 320 tons. Several trains only make three stops between Paris and Belfort.

From Paris to Nancy (353 km. = 220⅜ miles) the timing up to the summer of last year was 5 hrs. 15 min.; since which period, however, it has been reduced to 4 hrs. 15 min., with trains of 320 tons. Again, the journey from Paris to Charleville (244 km. = 152½ miles) which occupied 3½ hrs. has been reduced to 3 hrs.

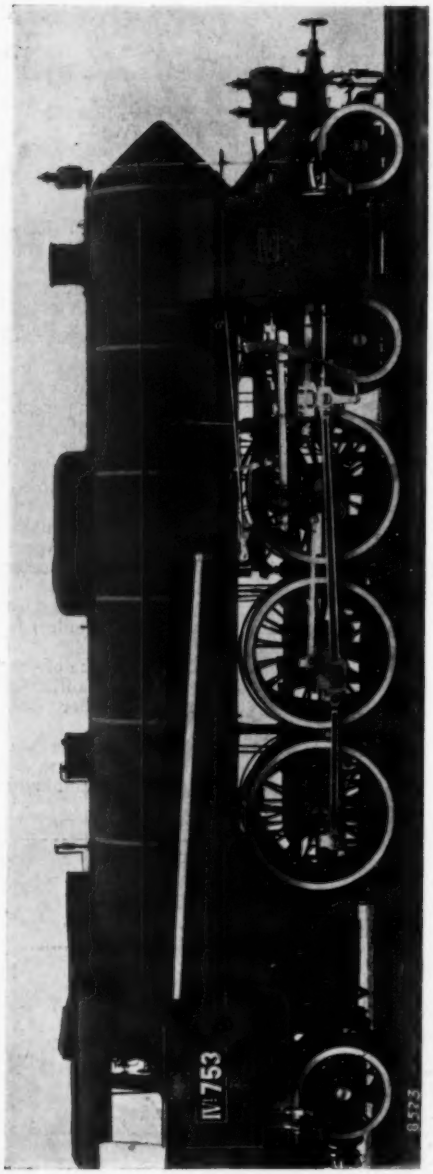
In order to keep these times and to make up for loss incurred by signal stops, permanent way slacks, &c., continuous speeds of 100 to 110 km. (62½ to 68¾ miles) must be maintained, although the maximum fixed by the State Control of 120 km. (75 miles) is frequently reached.

To give an idea of the excellence of the work done by these engines, M. Dantin, in *Le Génie Civil*, gives, amongst others, the following examples:

August 30th, engine No. 3112, train No. 30 with 400 tons behind the tender, made up 36 minutes between Chaumont and



4-6-0 TYPE FOUR-CYLINDER COMPOUND EXPRESS LOCOMOTIVE, EASTERN OF FRANCE RAILWAY.



4-6-2 "PACIFIC" TYPE EXPRESS LOCOMOTIVE, BADEN STATE RAILWAYS.

Paris over a distance of 262 km. (163¾ miles). October 23rd, train No. 28, with 460 tons behind the tender, made up between Nancy and Paris 15 minutes, notwithstanding the heavy load. We reproduce an outline drawing of an engine of the new type, of which the following are a few leading dimensions:

Cylinders H.-P. diameter	0·36 m.
,, L.-P. ,, ...	0·59 ,,
Stroke	0·68 ,,
Coupled wheels, diameter	2·09 ,,
Bogie ,, ,, ...	0·92 ,,
Heating surface, fire-box	16·22 sq. m.
,, ,, tubes	218·69 ,,
,, ,, total	234·91 ,,
Grate area	3·16 ,,
Boiler, mean diameter	1·55 m.
Tubes, number	140
,, exterior diameter	0·07 m.
,, length between tube plates	4·40 ,,
Weight empty	70·16 tons.
,, loaded	76·79 ,,
,, adhesive	53·24 ,,
,, including tender	115·24 ,,

New German "Pacific" Type Locomotive.

We have already illustrated one example of the "Pacific" type of locomotive now being introduced on the Continent and in Great Britain for the heaviest passenger express trains, and with larger boilers than it is possible to use in ordinary six-coupled engines. As announced in M. Demoulins' recent work on locomotives, the Baden State Railways was at that date building the first example of the "Pacific" type at the München Works of Messrs. J. A. Maffei & Co., and to be finished at the end of 1906; but the first engine was not delivered until August last.

It is notable as combining a boiler generating steam at 235 lb. working pressure with a steam temperature of up to 300° C., the engine proper being of the four-cylinder balanced compound type. It is significant that the same builders designed some previous engines with pressures of over 200 lb., and now have returned to higher pressures again, and presumably for reasons that have dictated the same practice in marine engines—low pressures involving proportionately larger boilers and more dead weight. The new engine is 30 to 50 per cent. more powerful and 20 per cent. heavier than the most powerful six-coupled two-cylinder low-pressure engines that are used on the Prussian State Railways. It is designed for hauling express trains on gradients of from 1 in 50 to 1 in 60 of over 20 miles in length, and to maintain a speed of

62 miles per hour with a train of 434 tons total, in ascending gradients of 1 in 300. The constructive details are: four cylinders and four piston valves all in one transverse line; all connecting rods driving on the same pair of wheels; one set of Walschaert's valve gear for the four valves; bar-frames throughout engine bogie-frames of plate; tender bogie-frames of diamond pattern. The engine never works simple expansion. Quick starting in all positions of the cranks is aided by an automatic admission of the steam through small inlet-valves fitted one at each extremity of both low-pressure cylinders and connected by a rod to the arms of the weigh-bar shaft as illustrated. The engine proper requires no more attention in operation than a four-cylinder simple expansion engine. Provision for water in the cylinders of this high superheat steam engine is made just as for saturated steam engines, as visible in the view. The superheater is of the U fire-tube type, now universally employed with other variants of the original French system.

The following are some leading particulars:

Grate area	48·4 sq. ft.
Heating surfaces, fire-box ...	172 ,, ,,
,, ,, tubes ...	2,195 ,, ,,
,, ,, superheater	430 ,, ,,
,, ,, total ...	2,797 ,, ,,
Steam temperature	300° C.
,, pressure	235 lbs.
High-pressure cylinders, diameter	16½ in.
High-pressure cylinders, stroke	24 ,,
Low-pressure cylinders, diameter	25½ ,,
Low-pressure cylinders, stroke	26½ ,,
Driving wheels diameter ...	70½ ,,
Engine weight	76 tons.
,, ,, loaded	85 ,,
Adhesion when loaded ...	48 ,,
Maximum load per axle ...	16 ,,
Minimum radial curves ...	160 m.
Maximum grade...	20 per cent.
Tender, coal capacity	7 tons.
,, water capacity ...	20 ,,
,, weight empty	21·8 ,,
Engine and tender loaded ...	134 m. tons.

"Brotan" Type Locomotive.

By the courtesy of the makers, Messrs. Beyer, Peacock & Co., we are able to present some particulars and illustrations of an interesting and powerful tank locomotive, recently completed by them for the British Mannesman Tube Company, Ltd., Landore, who intend placing it in shunting service. It is of powerful proportions, and has been designed to give the necessary adhesion and prevent slipping under the high tractive effort exerted by cylinders 14 in. by 20 in. with a pressure of 180 lb. The "Brotan" system, on which the locomotive is constructed, may not be altogether unfamiliar to English engineers, as for some time past locomotives of this type have been in service on the Austrian railways; but this, we believe, is the first example of the kind to be employed in this country, and is consequently not without interest. As will be gathered from the illustrations, the main feature of the system is the boiler, in which the ordinary inside fire-box is replaced by a system of water tubes, involving the elimination of the customary water space and stays. The boiler proper is divided into two cylindrical portions fixed parallel to each other, the lower being the main portion, and contains a number of fire tubes. Connection with the upper barrel is made by means of two necks. The upper ends

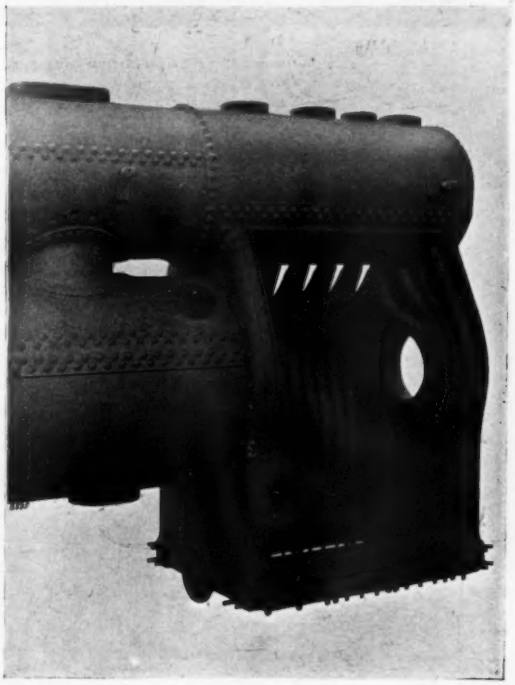


FIG. 1.—"BROTAN" TYPE LOCOMOTIVE, DETAILS OF WATER-TUBE FIRE-BOX.

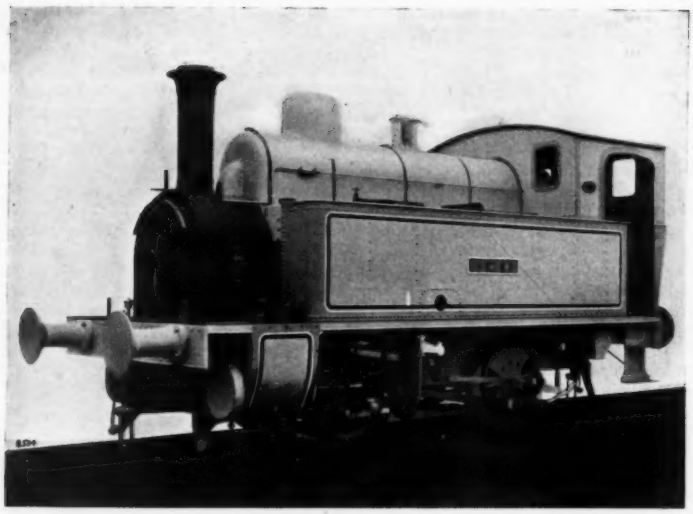


FIG. 2.—"BROTAN" TYPE LOCOMOTIVE COMPLETE.

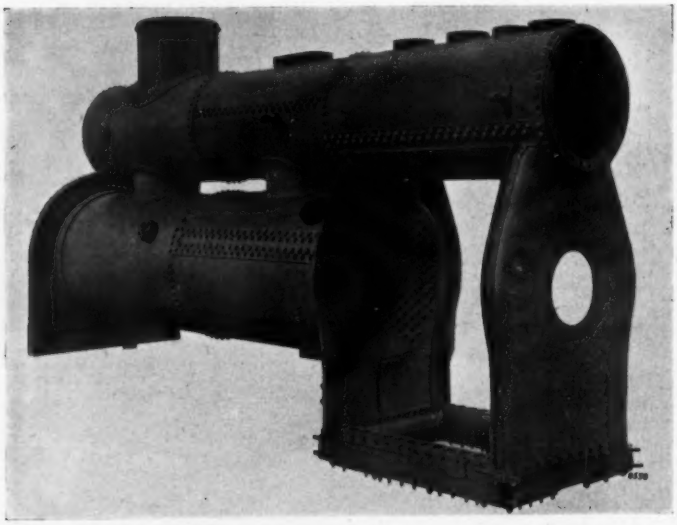
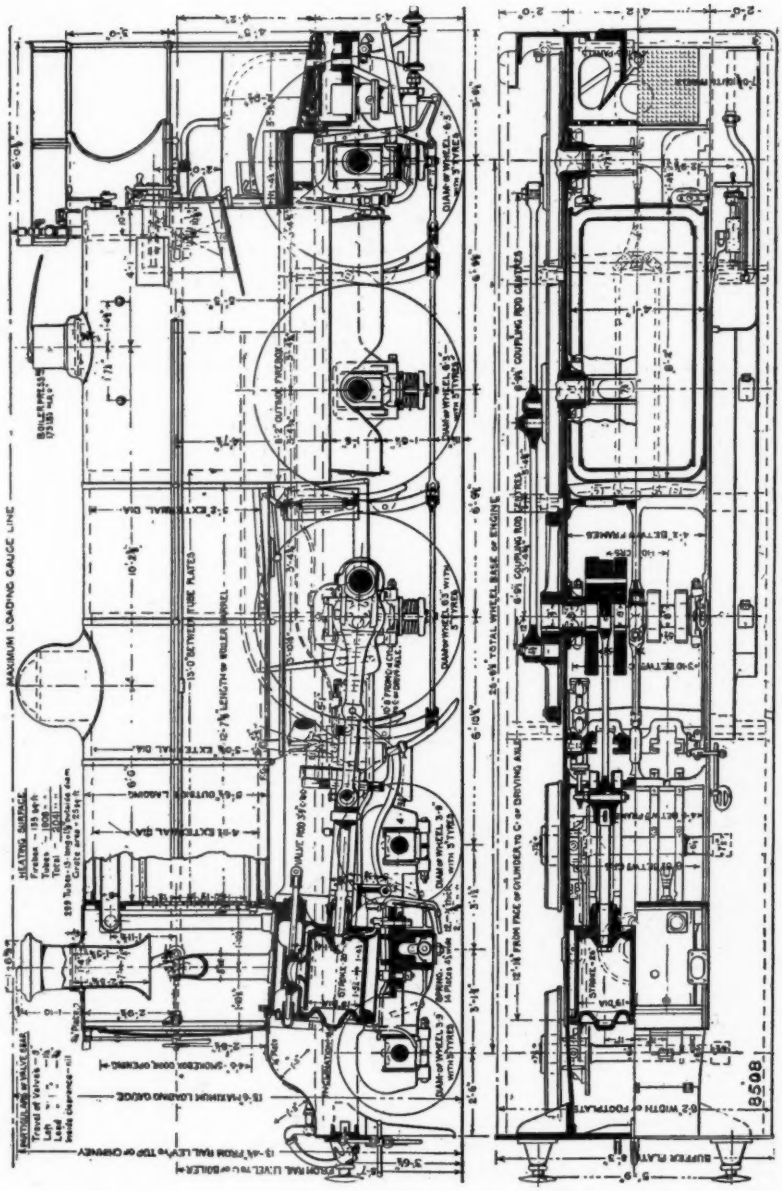


FIG 3.—VIEW OF THE "BROTAN" BOILER SHOWING THE ARRANGEMENT OF THE COMPONENT PARTS. THE WATER TUBES HAVE BEEN REMOVED FROM THE FIRE-BOX.

of the water tubes, composing the fire box, are fixed in the rear end of the upper barrel, which, for this purpose, is of thicker plate as shown. The lower ends of the water tubes are expanded into a rectangular shaped water circulating chamber of steel, connection between which and the back end of the main barrel is made by a large pipe, of which, however, only the flanges are visible in the photograph, Fig. 1. Facilities for inspecting and cleaning the water tubes are provided by means of a number of removable doors on the underside of the circulating chamber, giving access to corresponding holes into which the water tubes themselves are expanded. The construction of the fire box is well shown in Fig. 1, in which one row of tubes has been removed. The tubes are encased in fireclay and a covering of sheet steel plates. The



4-6-0 TYPE EXPRESS LOCOMOTIVE, "EXPERIMENT" CLASS, LONDON AND NORTH WESTERN RAILWAY.

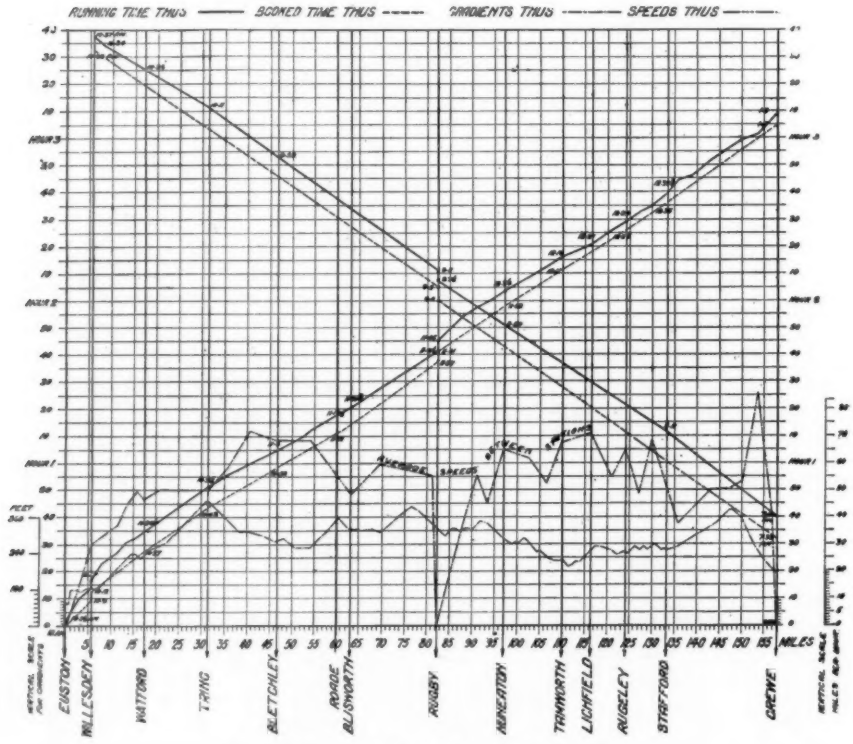
following are the leading dimensions of boiler and engine:

Cylinders, diameter	14 in.
,, stroke	20 in
Valve gear	Walschaert.
Wheels, diameter	3 ft. 1 in.
Wheel base	7 ft. 6 in.
Main barrel, inside diameter ...	3 ft. $0\frac{3}{4}$ in.
Top drum ,, ,, ...	2 ft. $3\frac{1}{4}$ in.
Length between tube plates... ...	9 ft. $6\frac{1}{2}$ in.
Number of smoke tubes	153
Outside diameter of smoke tubes ...	$1\frac{3}{4}$ in.
Number of water tubes	34
Outside diameter of water tubes ...	$3\frac{1}{4}$ in.
Fire-box, length outside	4 ft. $3\frac{1}{4}$ in.
,, width outside	3 ft. 3 in.
Height from top of circulating box to underside of top drum ...	3 ft. $3\frac{7}{8}$ in.
Heating surface, boiler	746 sq. ft.
,, ,, tubes	669 ,, ,,
,, ,, fire-box	77 ,, ,,
Grate area	9'2 ,, ,,
Weight in working order	30 tons 10 cwt.
Working pressure	180 lb.
Tank capacity	700 gals.

This mode of construction might appear to involve extra weight; as a matter of fact, the boiler is not quite so heavy as one constructed on ordinary lines.

Everyday Work of Leading British Locomotives: 1. The L. & N. W. Railway.

Since the inauguration of "Railway Notes" as a regular feature of this journal, an endeavour has been made to keep readers abreast of all important developments in locomotive design and construction.



RUNNING OF THE 10 A.M. TRAIN, EUSTON TO CARLISLE.

Euston to Crewe :—Engine, "Phalaris," 4-4-0 "Precursor" Type. Number of Vehicles, 12. Number of Axles, 54. Weather, foggy and damp.

Little attention has, however, been devoted to the work the engines themselves are being called upon daily to perform: yet, apart from anything of a sensational character which is of comparatively rare occurrence, there is enough interest attaching to the consideration of regular locomotive performance to justify some space being devoted month by month thereto.

In considering the winter services of 1907-8, a commencement may be made with the work of Mr. G. Whale's engines of the "Precursor" (4-4-0) and "Experiment" (4-6-0) types on the L. & N. W. Railway.

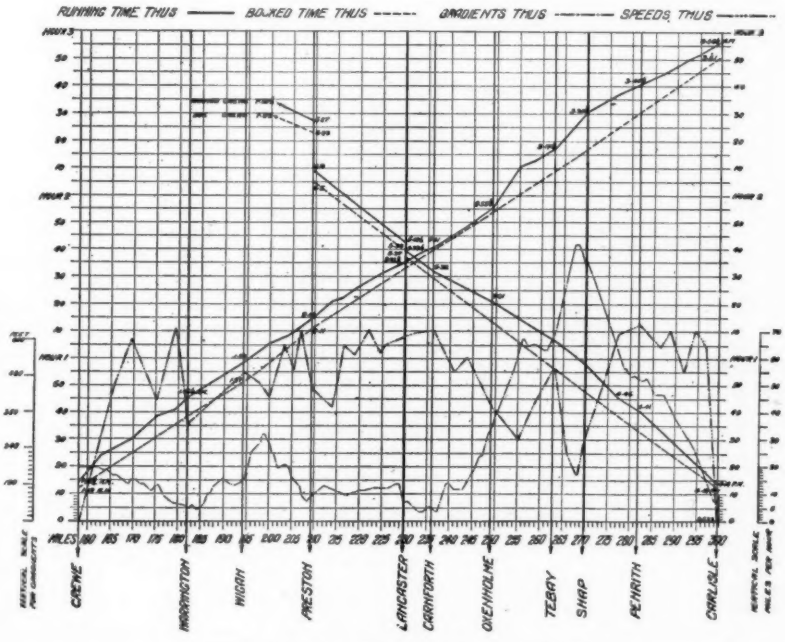
These now well-known series of engines, which were introduced in March, 1904, and May, 1905, respectively, are of straightforward design, and have earned for themselves, both as regards speed and efficiency, a most meritorious reputation Indeed, in actual work they are, in spite of the very heavy loads they are called upon to handle, the fastest machines ever built by the L. & N. W. Company.

The following are the leading dimensions:—

	"Precursors."	"Experiments."
Cylinders	19 in. × 26 in.	19 in. × 26 in.
Coupled wheels diameter	6 ft. 9 in.	6 ft. 3 in.
Heating surface	2,009·7 sq. ft.	2,041 sq. ft.
Grate area	22·4 ,,	25 ,,
Steam pressure	175 lbs.	185 lbs.
Adhesion weight ...	38 tons 15 cwts.	46 tons 15 cwts.
Total weight	59 ,, 15 ,,	65 ,, 15 ,,

On the run selected for description, viz., the 10 a.m. day Scotch Express, Euston to Glasgow, engines of both types were employed, thus affording a good opportunity of comparing their respective capabilities.

The train consisted of nine vestibuled bogie coaches and two dining cars with a total of fifty-four axles, the engine being "Phalaris," 1297, "Precursor" type. It is allowed 97 minutes from Euston to Rugby (82¾ miles) with a three minutes stop at Willesden Junction, and 84 minutes from



RUNNING OF THE 10 A.M. TRAIN EUSTON TO CARLISLE.

Crewe to Carlisle:—Engine, "Atalanta," 4-6-0 "Experiment" Type. Number of Vehicles, 16 Number of Axles, 70 Weather damp.

Rugby to Crewe (75¼ miles). The weather being foggy, the departure from Euston was at 10 hr. 2 min. 37 sec. or 2½ mins. late; and, on leaving Willesden, five minutes had been lost. From the latter place Watford (12 miles) was passed in 17 min. 10 secs. and Tring (26¼) in 34 min. 15 secs., a steady speed of 50 m.p.h. being maintained up the 1 in 330 rise between King's Langley and the Summit. On the run down to Wolverton a rapid acceleration took place culminating in a maximum of 73 m.p.h. Bletchley (8½ miles) from Tring was passed in 7 min. 55 secs., the speed falling, however, to 47 through Blisworth, Rugby being finally reached at 11 hr. 41 min. 20 sec., slightly under schedule, although actually 4 min. 20 secs. late.

Rugby was left at 11.45 a.m. or 4 mins. late, the 14 miles to Nuneaton occupying 18¼ mins., and the 19 miles from the latter place to Lichfield 18 min. 35 sec., the speed again touching 72 m.p.h. on the descent of 1 in 330 outside Tamworth. A careful slack round the curve into Stafford brought the passing time to that station to 54 min. 35 sec. from Rugby (50¾ miles). The subsequent rise to Whitmore was taken at 50 m.p.h., soon, however, increasing on the down run to Crewe of 1 in 177 and 1 in 230, to a maximum of 85 m.p.h., the train coming to rest at Crewe at exactly 1 hr. 9 min. or 4 mins. late. The 75¼ miles from Rugby thus occupied 84 mins.—schedule time—an average of 53½ m.p.h.

The fluctuations in speed characterising this run will best be followed by the aid of the diagram on p. 174, on which curves are plotted of running time, booked time, gradients, average speeds between stations, affording an opportunity of an instructive comparison. The method of plotting the time curve needs explanation, as the peaks of the curve do not necessarily show that the maximum speeds occurred *exactly* at the point indicated: the coincidence is, however, sufficiently near for all practical purposes. Time observations were taken when passing each station or important junction, &c., and the average speed computed therefrom between each observation and spotted on the chart, the curve being drawn through all such spots.

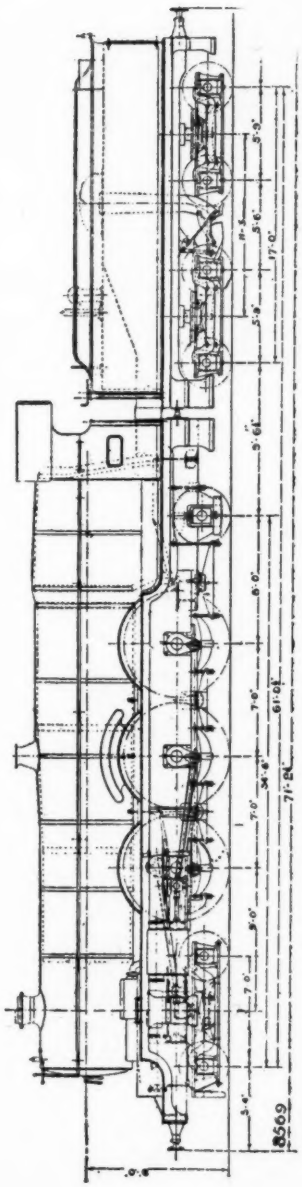
Resuming the details of the run: at Crewe the load was increased by the Birmingham portion coming on, consisting of four bogie vehicles, including the through Bristol - Glasgow carriage which had travelled *via* the Severn Tunnel. The total load thus became = 70 axles, which was taken by "Atalanta," No. 61, "Experiment" class, the departure being made at 1.13¾ min.,—1¾ mins. late. The train is allowed 159 mins. for the 141 miles separating Crewe from Carlisle, representing an average speed of 53 m.p.h. This is one of the longest non-stop runs in Great Britain, and, considering the heavy climb over the Shap, one of the most meritorious. A bad permanent way slack soon after leaving Crewe caused the 5 miles to Minshull to occupy 9 mins. 27 secs.; speed, however, soon rose to 67 increasing to 70 through Moore, after which another slack reduced the running to 4½ mins. for the 2¾ miles to Warrington. Wigan, 36 miles from Crewe, was passed in 44¼ mins., the run up to Coppull at 1 in 100 and 1 in 330 being taken at 50 and 46 m.p.h. Down the bank 74 was reached through Farington, Preston, 51 miles from Crewe, being passed 1 hr. 40 secs. out. On the comparatively level stretch to Lancaster, speeds of 62 and 73 were reached, the run down to Carnforth being maintained at a steady 70. The 23¾ miles from Barton, just outside Preston to Carnforth, occupied 20 mins. 15 secs. Milnthorp (85½ miles) was passed in 94½ mins. from the Crewe start, and the 12¾ miles to Grayrigg, chiefly at 1 in 133 and 1 in 100, accounted for 21 mins. 50 secs. owing to a slack. Tebay (104¼) was passed in 123 mins. 17 secs. from Crewe, the speed up the long climb of 1 in 75 to the Summit falling to 25 and 17¾ m.p.h. The run down to Carlisle from Shap Station 29¼ miles was made in 25¾. mins., the train coming to rest at Carlisle at 3 hrs. 56½ mins.—5½ mins. late. Eliminating time lost on account of slacks, however, schedule time was again kept.

Beyond the high speed (86) reached between Madeley and Betley Rd. there was nothing to mark the run as being in any sense exceptional; it serves, however, as a good example of the fine work that is being consistently performed by Mr. Whale's locomotives. The return journey was made on the afternoon up Scotch Express, the principal timings only of which are plotted on the accompanying diagrams. In conclusion our thanks are due to Mr. Whale for courteously providing travelling facilities and data which assisted in the compilation of the diagrams.

4-6-2 "Pacific" Type Locomotive, Great Western Railway.

It has been known for some time past that the Great Western Railway have had

under construction in their shops at Swindon a locomotive of the "Pacific" type (4-6-2). The engine, No. 111, to which the name the "Great Bear" has been given, is now finished and at the time of writing is undergoing her trials. Her appearance in actual service is an event the importance of which will not be underestimated in railway circles as she is the first of the "Pacific" type to be introduced in this country, and therefore possesses peculiar interest. The engine is the outcome of a previous design—the "Star" class, possessing in common therewith the same cylinder, steam distributing and driving wheel arrangement. As compared with the "Stars," however, the cylinders are increased from 14½ to 15 in. diameter, the stroke remaining the same, viz., 26 in. The fixed wheel base has, however, been reduced to 14 ft., the coupled wheels themselves having been brought forward of the fire-box. This has removed the disability attaching to the 4-6-0 design, wherein the lateral fire-box dimensions are restricted by the rear coupled axle, an objection which also applies to the 4-4-0 class. Mr. Churchward has been able to obtain a fire-box heating surface of 158 sq. ft. and a grate area of 47 sq. ft., an increase of 4 and 15 sq. ft. respectively over the "Star" type. Support to the fire-box is afforded by a pair of radial carrying wheels, 3 ft. 8 in. diameter. It is expected that the shortened fixed wheel base and the provision of ample side-play at the radial wheel will enable the engine to easily negotiate any curves met with. The boiler is of the "Belpaire" type, having 2,242 sq. ft. of tube heating surface, which, together with the increased grate area, provides the high steam generating capacity necessitated by the use of 4-h.p. cylinders. Two of the latter are set inside the frames in advance of the bogie centre and drive the leading pair of coupled wheels, two other cylinders being arranged outside the frames in rear of the bogie centre and drive the middle pair of coupled wheels. The cranks are set on the four quarters, the outside crank pin on each side being at 180 deg. to its corresponding inside crank pin. The right-hand crank pin leads. Piston valves and "Walschaert" gear have been adopted, but only two gears, placed between the frames, are employed for operating the four valves. The combining lever is actuated by a guiding link from the inside crosshead, and the eccentric is placed on the crank axle. The valve of the outside cylinder is actuated by a horizontal rocking lever, worked from the valve rod of the corresponding inside cylinder. Reversing



4-6-2 "PACIFIC" TYPE EXPRESS PASSENGER LOCOMOTIVE, THE "GREAT BEAR," GREAT WESTERN RAILWAY.

is accomplished by screw gear, which, by means of an arm on the reversing shaft, raises or lowers the radius rod in the link, through the intervention of a Scotch yoke, or slotted crosshead, formed on the radius rod. The boiler is of the "Belpaire" type, with wide firebox, and is placed with its centre 9 ft. above rail level. The barrel is 23 ft. long and 6 ft. maximum diameter, tapering to 5 ft. 6 in. at the smokebox. The firebox, which is sloped backward to clear the trailing coupled wheel, is 8 ft. long and 6 ft. 6 in. wide at the foundation ring, tapering upwards at the sides to 5 ft. 9 in. It is fitted with a "Swindon" type superheater. The firebox brick arch is carried on four arch tubes 3 in. internal diameter.

The following is the detailed heating surface :—

	Sq. ft.
141 flue tubes 2¼ in. outside dia., 22 ft. 7 in. long	2,673·45
21 " " 4½ in. " " " "	
84 super " 1⅝ in. " "	545
4 arch " 3⅛ in. " "	24·22
Firebox	158·14
Total	3,400·81
Grate area	41·79

The boiler is bolted at the smokebox to the cylinders and the saddle casting, and is carried at the firebox by a special sliding bracket at the front and a plate bracket at the back to allow for expansion, which in a boiler of these dimensions is considerable. The bogie is of the new design with spring side control, the weight of the engine being taken by two spherical cups, bearing on flat sliding surfaces on a stiff centre casting, the ordinary centre pin being retained for controlling purposes. The spring gear is equalised from the leading coupled wheel to the trailing carrying wheel. The vacuum brake is fitted to all wheels of engine and tender except the trailing carrying wheels. The working pressure is 225 lbs. per sq. in., and the tractive effort at 90 per cent. equals 29,430 lbs. The tender differs from the standard G.W.R. tender in that it is carried on two bogies. These bogies are identical with the engine bogie except that the wheel base has been shortened to 5 ft. 9 in., while that of the engine bogie is 7 ft. The tank is of 3,500 gallons capacity, and is fitted with an improved form of balanced water pick-up. The coal capacity is 6 tons. The wheel base of the engine is 34 ft. 6 in., and that of the tender 17 ft., and the total overall base of engine and tender is 61 ft. 0½ in.

In working out the design it will be noticed that the standardisation and interchangeability of detail parts has been carried out as far as possible. As already mentioned, the bogies of the engine and tender are identical except for wheel base, the wheels and axleboxes, horns, springs, &c., being interchangeable. These details are also standard with the bogies and pony trucks now running under 4-4-0, 4-4-2, 4-6-0, 2-8-0, $\frac{2\text{-}6\text{-}2}{T}$, and $\frac{4\text{-}4\text{-}2}{T}$ engines.

The coupled wheels and axles are interchangeable with those of the "Star" class, as also are the connecting rods, valves, valve gear and reversing gear. The axleboxes and springs are interchangeable with all engines of the above classes, and the trailing wheel and axle is interchangeable with $\frac{4\text{-}4\text{-}2}{T}$ and $\frac{2\text{-}6\text{-}2}{T}$ classes. The following are the leading dimensions of the "Great Bear" and "Star" classes :—

Class.	"Great Bear." 4—6—2	"Star." 4—6—0
Wheels—	ft. in.	ft. in.
Driving	6 8½	6 8½
Bogie	3 2	3 2
Trailing...	3 8	—
Wheel base—		
Centre of bogie to leading...	9 0	9 0
Driving...	14 0	14 9
Trailing to Carrying	8 0	—
Cylinders (4)—		
Diameter	0 15	0 14½
Stroke	0 26	0 26
Boiler barrel—		
Length	23 0	14 10
Diameter (maximum) ...	6 0	5 6
Heating surface—	sq. ft.	sq. ft.
Tubes	2,673	1,989
" (arch)	24	—
" (superheater)	545	—
Firebox	158	154
Total	3,400	2,143
Grate area	42 sq. ft.	27 sq. ft.
Working pressure (per sq. in.)	225 lb.	225 lb.
Tractive effort	29,430 lb.	26,560 b.

Owing to the fact that the photograph of the engine was only received at the moment of going to press, we are compelled to postpone reproduction until the next issue.

LEADING ARTICLES OF THE MONTH—CURRENT EVENTS—TECHNICAL INFORMATION FROM ALL SOURCES.

Roller Bearings.—At a recent meeting of the Institute of Marine Engineers, Mr. Geo. B. Woodruffe gave an interesting lecture on this subject. In opening, the lecturer contrasted the three distinct types of anti-friction bearings: ball, solid and flexible roller bearings. The steel ball, he said, was a most desirable type for light loads, but for heavy loads, on account of having only points of contact, abrasion of races and breaking of balls is frequently the result. If very large balls were used then the bearing became very expensive. They must be hardened and ground and required surfaces similarly prepared, and were liable to fracture due to sudden shock and defective temper. Regarding the solid roller, on account of the rigidity apparent from its construction, it could not, in its practical operation in a bearing, have contact along its entire length. It was impossible, therefore, to secure a uniform distribution of load, either on the face of the roller or the axle or the shaft. It was applicable to heavy loads principally, there being more contact with the supporting surface than with the ball, but owing to the rollers being rigid and unable to conform to irregularities always present, the support for loads consisted of a series of points, not a line. It was not suitable for high speeds and was liable to fracture as in the case of the ball. The flexible roller bearing, a spirally-wound roller of flat bar steel forming a coil or spring of uniform diameter, on account of its construction presented at all times a bearing along its entire length, resulting in a uniform distribution of load on the roller itself as well as on the surfaces on which, and in which, it operated. It was applicable to all speeds and loads, due to ability to vary its nature, the rollers acted as oil reservoirs, and the right and left spirals as oil carriers, the necessity of hardened and ground steel sleeves was eliminated, and there was a less co-efficient of friction. This type had not been applied to any great extent in marine work, but he estimated that a combination of flexible roller bearings on the tunnel shaft and the new "Mitchell" thrust bearing to take the place of the old-fashioned thrust block, would bring about a great reduction in the coal consumption. In reply to questions, Mr. Woodruffe said that roller bearings were not suitable for the crank-shaft of reciprocating engines, but were especially suitable for tunnel shafting. He would not recommend their use for motor engines. The steel used for the bearings was not hardened. The effect of heavy pressures upon the spirals was inappreciable.

Stresses in Masonry Dams.—Three important papers* on this subject were recently read before a meeting of the Institution of Civil Engineers, two of which embodied results of investigations carried out with "plasticine" and india-rubber models that were of a considerably novel and interesting character.

The experiments described in the first paper were carried out in the shops of the late Royal Indian Engineering College at Cooper's Hill, and occupied from first to last a period of about fourteen months. They were restricted, for reasons given, to models of a dam of typical triangular section under perfect conditions, and the results obtained are submitted as a contribution to existing knowledge on the important subject of the arrangement of stresses in masonry dams subjected to water-pressure. The models were made of "plasticine," a kind of modelling clay, which appeared likely to reproduce on a small scale many pf the conditions existing in a full-size structure. The specific gravity of this material is 1·33; its temporary elastic limit in tension is about 2 lbs. per square inch, whilst its temporary elastic limit in compression is about 3 lbs. per square inch. The word "temporary" is used in this connection because after a time the plasticity of tbe material obscures its elastic tendency. It was also found that "plasticine" possesses a property of peculiar value in such experiments, in that, notwithstanding its plasticity, it will crack quite short along the plane of maximum

* "Experimental Investigations of the Stresses in Masonry Dams subjected to Water Pressure," by Sir John W. Ottley, K.C.I.E., and A. W. Brightmore, D.Sc., MM.Inst.C.E.; "Stresses in Dams: an Experimental Investigation by means of India-Rubber Models," by J. S. Wilson and W. Gore, Assoc. MM.Inst.C.E.; and "Stresses in Masonry Dams," by E. P. Hill, M.Inst.C.E.

tension when subjected to relatively small tensions for a sufficiently long period. The dam was first modelled of triangular section with the vertical face exposed to the pressure of the water, the base being made equal to the height divided by the square root of the specific gravity of the "plasticine," so that the resultant of the pressure on the base (due to the weight of the model dam itself and the pressure of the water) would act at one-third of the width of the base from the outer toe. The height of the model was taken at 30 in., which therefore gave a base of 26 in. The length of the dam was 12 in. In subsequent experiments the width of the base was reduced to 23, 20 and 17 ins., whilst in one case a sloping face was adopted. The model was moulded in a frame furnished with thick plate-glass sides, and, in order to permit of accurate measurement of any displacement in the model, corresponding horizontal and vertical lines, 2 in. apart, were scratched on both the glass and the "plasticine." The glass sides were made adjustable, and, previous to the application of water-pressure, care was taken to see that the lines on the glass coincided accurately with the corresponding lines on the "plasticine." A clearance was left between the glass sides and the model, to ensure that the latter should receive no support from friction between it and the glass. Water-pressure was applied to the face of the model by water contained in a thin rectangular india-rubber bag made to fit the frame. The weight of the model as first made (with a 26 in. base) was 230 lbs., the water-pressure on the vertical face being 195 lbs. This gave an average value for the intensity of normal pressure on the base of about 0·74 lb. per sq. in., and an average intensity of shearing stress over the horizontal base of about 0·62 lb. per sq. in., thus keeping well within the elastic limit of the "plasticine." The displacements of the originally vertical and horizontal lines were shown in the various diagrams submitted. The first of these represented the condition of affairs after the water-pressure had been left on for one day. After these measurements had been taken, the water was syphoned off, and it is worthy of notice that in the course of a few days the model entirely recovered its original form, thus showing that the material was elastic under the pressure to which it had been subjected for the comparatively short period of one day. More important results were obtained when the water-pressure was left on for a longer time so as to allow the model to become permanently deformed owing to its plasticity, as it thus became possible to show more definitely the distribution of the stresses in the structure. A diagram showed the state of the model after the water-pressure had been left on for 33 days, a crack at the inner toe having commenced to appear after seven days. After detailing the method of calculation adopted, it is pointed out that both the recorded displacements of the lines on the model and the calculations based on these indications show that, although the old assumption that the shearing stress is proportional to the normal stress does not hold good at the *base* (because the dam is "fixed" at that level), yet at *higher levels* it is, approximately, correct, and that, therefore, the shearing stresses to be provided against are not those at the base, but at rather higher levels near the outer profile. It is further pointed out that although a dam may be designed so as to have no tension on the horizontal plane at the inner toe when subjected to water-pressure, still there are tensions on other planes passing through that point, and it is suggested that the plane of maximum tension at the inner toe can be made more vertical by tying the front part of the dam into the foundation below the level of the base. The permissible values of compressive and shearing stresses in dams are then considered, and the following final conclusions are drawn:—(1) If a masonry dam be designed on the assumption that the stresses on the base are "uniformly varying," and that these stresses are parallel to the resulting force acting on the base, the actual normal and shearing stresses, on both horizontal and vertical planes, would (in the absence of stresses due to such factors as changes in temperature, unequal settlement, &c.) be less than those provided for; (2) There can be no tension on any plane at points near the outer toe; (3) There will be tension on planes other than the horizontal plane near the inner toe; the maximum intensity of such tension being generally equal to the average intensity of shearing stress on the base, and the inclination of its plane of action being about 45 deg.

In the introduction to the second paper, reference is made to the mathematical and experimental investigations of the problem already made by Mr. Levy, Prof. Karl Pearson, and Mr. Atcherley. The description given by Sir Benjamin Baker, at the Institution, in the discussion on the Coolgardie water supply, of experiments he had made with jelly models of dams, suggested to the authors the possibility of making exact experiments with models of india-rubber, in place of jelly.

The paper is divided into sections, as follows:—

Method of Straining the Model and Construction of Apparatus.—The two india-rubber models experimented with are 1 in. thick and to a scale of $\frac{1}{240}$ represent sections of dams cut off by planes 20 ft. apart. The profiles of both models are the same in the upper parts, but at the base, section *B* joins the foundations by curves, and section *C* by sharp angles. In each case the dam-like portion is about 6·3 in. high, and the foundation portion is 4 in. high and 12 in. long. Each model is divided into imaginary prisms, and from the centre of gravity of each prism a weight is suspended, the virtual specific gravity of the whole model being thereby increased to about 40. The water-pressure acting on the up-stream face is reproduced by plates pulled against the model by

cords attached to weights, calculated so as to represent a fluid having a density of $\frac{40}{2{\cdot}25}$. The apparatus is arranged so that the weights, representing the masonry and water-pressure, can be supported by the model or by an independent frame. The models are made of such shapes that when fully strained they assume the profiles as designed.

The Elastic Properties of Rubber.—Attention is called to the fact that to take up the greater part of the strain corresponding with a given stress, rubber requires a period of about 24 hours. To regain its original form a similar period is necessary. The relationship between direct stress and the strains produced in three directions, for the quality of rubber used, was tested by an apparatus described in the paper. The results of tests are plotted as curves, which prove that the strains are directly proportional to the stresses within the limits of the tests.

Measurements of the Strains.—The strains are measured by the aid of photography. A pair of photographs are taken of the model, one when it is strained, and the other when it is unstrained. Black lines ruled on the light surface of the rubber appear on the photographic negatives as clear lines on a dark ground. The distances and angles between these clear lines are accurately measured with specially constructed scales and micrometer-gauges.

The Design of the Sections.—The dam-like portion of each section was designed to represent a dam 125 ft. high, and the profile was calculated, according to the best principles so far suggested, to fulfil certain requirements with reference to the absence of tensile stress and the maximum compressive stresses.

Results obtained.—Three sets of experiments were carried out with the models, and the stresses determined in each case are indicated in diagrams of ellipses of stress and three sets of curves, two giving the distributions of normal pressure on horizontal and vertical planes, and the third giving the distribution of shearing stress on one of those planes. The authors draw attention to the principal features revealed by the investigations.

Conclusions.—The following are some of the conclusions given in the paper:—(1) Tensile stresses may exist at the up-stream toe of a dam, notwithstanding the fact that the line of resistance lies well within the middle third. The tension may be reduced by (*a*) making the up-stream face vertical, or by otherwise increasing the weight of the dam toward that face; this would have the effect of increasing the stresses in the dam when the reservoir is empty; (*b*) by a general increase in the dimensions of the dam; (*c*) by placing an earth embankment against the down-stream face. (2) The direct stresses at the down-stream toe are compressive in every direction, but reduce to zero in the direction normal to the face. (3) The maximum compressive stresses in a dam above its foundations are in a direction approximately parallel with the down-stream face and generally some distance therefrom. In magnitude they are slightly greater than

$$\frac{P_T}{\cos^2 \phi}$$

where P_T is the maximum normal pressure on a horizontal plane as determined by the trapezium law, and ϕ is the angle between the resultant and the vertical. (4) The shearing stresses are considerable at or near the up-stream toe. They are a maximum a short distance from the down-stream face, in a plane approximately at 45 deg. to the face. The maximum shearing stresses are in magnitude equal to

$$\frac{P_T}{2 \cos^2 \phi}$$

(5) The stresses in the foundations are of less consequence than in the dam above the base, because of the lateral support and the more extended distribution. (6) The stresses are considerable at the toes of a dam if they form sharp angles with the foundations. These stresses may be reduced by replacing the angles with curves of large radii. The curve at the up-stream toe may take the form of a rounded quoin, cut in large stones, so as to avoid joints, in the masonry, normal to the direction of the greatest tensile stress.

In the third paper the method of determining stresses starts by assuming that the vertical pressure on the base varies uniformly from one side to the other. From this assumption principal stresses are derived, the magnitude and direction of which vary considerably from the stresses calculated by the method hitherto adopted. The stresses in the Periyar dam have been investigated by the method given in this paper, and were shown in a diagram.

Notes on the Erection of Cantilever Bridges.*—This paper deals with the erection of cantilever bridges by the process of corbelling forward, and the temporary stresses which take effect when the process is extended to the central "independent" span, by using its panels as a temporary prolongation of the cantilever. During this part of the process the stresses due to the weight of the structure are greatly changed—not only in the members of the independent span, but also in the cantilever. The river-arm and also the shore-arm of the cantilever are now subjected to greater bending moments and greater boom-stresses, and these are briefly referred to; but the more important changes are those which will generally take effect in the web-system of both arms of the cantilever, and they become especially important when the web-system consists of vertical posts and diagonal ties designed to act only in tension.

Taking any ordinary form of cantilever, of varying depth, it is shown that, while the

* Abstract of a paper by Prof. T. Claxton Fidler, M.Inst.C.E., read before the I.C.E., Feb. 11th.

vertical shearing force at all points in the river-arm undergoes no change under the altered condition of the structure, yet the tensile stress in the diagonal of every *tapering* panel is considerably *less* than the stress due to the same load in the completed bridge. Thus, it is possible that the tensile stress in one or more diagonals may fall to zero as soon as the corbelling process reaches a certain definable stage, and if the process goes further the stress may be reversed in direction. Any such reversal of stress would tend to induce a buckling of the lower boom, and it is, therefore, necessary to examine this contingency in detail.

To define and to simplify the problem, the cantilever is first treated as a pin-connected frame of discontinuous bars, free to turn at the joints (the diagonal ties being flexible). The arm, which projects beyond any given panel, can then be examined in respect of its equilibrium while the process of corbelling goes on; and the stability of the arm vanishes as soon as its centre of gravity reaches a certain critical point *P*. If the corbelling process is carried any farther, the collapse of the pin-connected frame follows as a necessary consequence from the upward buckling of the lower boom at one or more of its joints.

The critical point *P* is determined for each panel by a simple graphic method, and serves the purpose of a metacentre, so that the question of stability can readily be determined when the centre of gravity has been found.

Passing from the hypothetical illustration, the lower boom is next considered as a continuous member following the same general outline; and the upward buckling tendency is most clearly apparent when the member is designed to follow an arched outline (convex upwards). The continuity of the member does not greatly alter the governing conditions as found in the pin-connected frame. At any stage in the corbelling process the same forces are in operation producing the same buckling tendency; and as the centre of gravity moves out to the point *P*, the lower boom passes under a new set of conditions. It first loses the support of those forces or reactions on which it had depended for its stiffening in the vertical plane. Then, as the corbelling goes on, the boom is subjected to positive transverse bending forces or stresses with an ever-increasing tendency to push forward the upward buckling movement.

The shore arm of the cantilever is to be treated in the same way, the anchorage force being duly taken into account; for in this arm also it is equally necessary to consider the possible contingency of an upward buckling of the lower boom under like conditions of equilibrium.

The general practical question whether the corbelling process can be carried out so far as to reach the centre of the span (or to reach any given intermediate point), without involving this contingency in either arm of the cantilever, must depend in each individual case upon the features of the design—the length and weight of the independent span, the geometrical form of the cantilever, and the curvature of its lower boom—and also upon the actual distribution of the load upon every panel of the bridge.

Test of a Live-Steam Feed-Water Heater.*—The authors of this paper have long wished for an opportunity of making an exhaustive test of the efficiency of a Live-Steam Feed-Water Heater. Many of the tests which have been made appear to show that some economy results from the use of such heaters, but it is very difficult to see the reason why economy results from taking live steam from a boiler and returning it immediately afterwards when mixed with the feed-water. In general, the percentage of saving claimed is not great, and it is evident that in order to get a really reliable comparison of the efficiency of a boiler with and without a live-steam feed-water heater, extreme care must be exercised in making the tests, and moreover, the conditions under which the plant is working during the comparative tests must be uniform. Such an opportunity presented itself at some works in Leeds, where the conditions for testing are quite ideal, since the output of the boiler is as nearly constant as it is possible to be, owing to the fact that the work on the engine does not appreciably vary from day to day.

The boiler under test was of the "dry back" type made by Messrs. Davey, Paxman & Co., Chelmsford.

The feed-water is supplied by means of a force-pump driven from the cross-head of the main engine; on its way to the boiler it passes through a filter for removing oil, and thence (when in use) through a live-steam feed-water heater.

The results of the tests show that the boiler evaporated almost exactly the same quantity of water per lb. of coal both when the heater was in use and when not in use, the results being even slightly better when the boiler was working without the heater, the actual increase in efficiency being 0·6 per cent. It would be useless to pretend to this degree of accuracy in any boiler trial, and the difference might easily be accounted for by errors of observation, notwithstanding the great care taken to eliminate all errors, but at the same time the authors would point out that with the heater in use the actual heat radiation surface was increased by that of the heater, and if this be taken into account the two tests agree even more closely. They are therefore forced to the conclusion that the heater has no material effect on the economy of the boiler, but from all appearances the deposit in the boiler is materially less when the heater is in use than when the feed-water passes direct to the boiler.

* Abstract of a paper by Prof. John Goodman, member, and Mr. D. R. Maclachlan, of the University of Leeds, read before the I. Mech. E., Feb. 21st.

ELECTRICAL PROGRESS

By ANDREW STEWART, A.M.I.E.E.

Railway Electrification Problems.

The past few weeks have been prolific in producing discussions and papers relating to electric transport. Perhaps the most noteworthy has been the valuable contribution which Prof. Gisbert Kapp has made to the subject by his lectures to the Royal Institution of Great Britain. It was Prof. Kapp who, more than anyone else, placed dynamo and motor design on a firm foundation by his lucid treatment of the subject in "Electric Transmission of Energy," a work which first saw light some twenty-two years ago. His lectures to the Royal Institution on railway electrification should prove as valuable to railway engineers, as was his historic work of two decades ago, to electrical men.

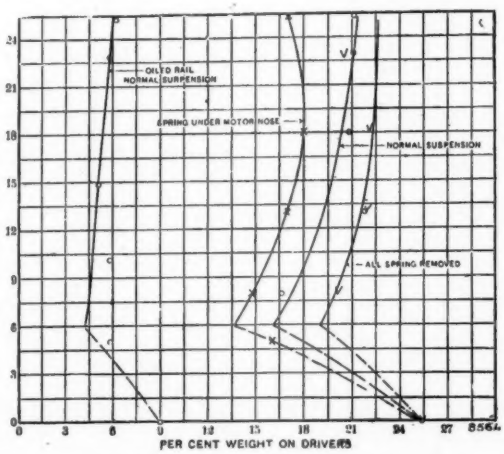


FIG I. CO-EFFICIENT OF ADHESION FOR ALTERNATING CURRENT MOTORS.

Early in his first lecture, Prof. Kapp emphasised the points which we have from time to time impressed on our readers, viz., the ability of electric trains to deal with heavier trains at higher speeds than steam: the power of a steam locomotive is limited by its ability to generate steam, and is dependent on the quality of the fuel supplied to it.

It is altogether different with the electric locomotive which has all the resources of a large power-house of many times its own capacity to draw upon. Such a power-house is not so susceptible to variations in the quality of fuel as a locomotive boiler, often worked to the last pound of its evaporative capacity. Moreover, the electric locomotive is not dependent on the power it can generate, its output is limited by what it can absorb, and this may be several times its rated maximum for short periods. It is also capable of sustained effort to an extent which is hardly appreciated by the engineer whose experience has hitherto been limited to a power generating plant on wheels, which is, after all, the correct description of a steam locomotive. All these features were enlarged upon by Prof. Kapp in a manner which does not permit of satisfactory condensation, and we would advise our readers to carefully peruse the lectures.

As a contribution to a question on which much diversity of opinion exists, Mr. A. J. F. Armstrong's paper on "The Choice of Frequency for Single Phase Alternating Current Railway Motors," read to the American Institute of Electrical Engineers a short time ago, is interesting. It will be seen from the curve reproduced here, Fig 1, that the co-efficient of

adhesion is greatest with alternating current, which of course is well-known, but that it should be a minimum at 6 cycles, and increase above or below this has not hitherto been recognised. The fact that its torque is a fluctuating one, and therefore the available tractive effort for a given weight on the drivers is not quite so great as with direct current, has always been appreciated, but there has been no accurate data to enable one to assess the difference. These curves show that while the difference is in some cases considerable, better results can be obtained by modifications of the suspension, but the point brought out by Mr. Armstrong is the small difference between 25 and 15 cycles. As our readers are aware, this question of best periodicity has been discussed from many standpoints, but the point now cleared up is that between 15 and 25 cycles there is no appreciable difference, and if standard 25 cycle plant can be used for railway electrification schemes, there is no reason whatever why non-standard, and therefore more expensive plant, should be introduced in order to get 15 cycles.

A point not discussed by Mr. Armstrong is the influence of still higher periodicities; from the contour of two of the curves we should be justified in assuming that 50 cycles will be very little less satisfactory than 25 cycles, and as 50 cycles is a common standard in this country and on the Continent, and moreover, the sub-station plant will be much cheaper at this than at any lower periodicity, there seems no reason why the higher periodicity should not get wider recognition from American engineers than it now does.

On the Continent, and to a certain extent in this country, it is recognised that 50 cycle single-phase motors are quite satisfactory, moreover it enables very much cheaper transforming plant to be used, and station and train lighting is very satisfactory. Beyond 50 cycles little standardisation has been effected in Europe, and the single-phase motor becomes much less satisfactory in performance.

The American engineers whose standards are 60 cycles and 25 cycles, feel that unless the new standard of 50 cycles is adopted they must fall back on 25 cycles. As electric railway practice in America is largely dominated by manufacturing interests, it is likely that 25 cycles will become standard single-phase railway practice, but English railway engineers would be well advised not to slavishly imitate American practice in this respect. Indeed it is one point where the excessive standardisation of the American tells against him. When once an American manufacturer has set a standard he is more conservative than an Englishman; he will not only attempt to work his standard practice into cases more or less suitable for it, which is the legitimate field of standard apparatus, but he will frequently accept serious handicaps in other directions, rather than depart from the lines laid down as a matter of manufacturing expediency.

Another, and a no less interesting view of the railway electrification problem, is presented by Mr. Bela Valatin in a recent issue of the *Street Railway Journal*. This gentleman takes one of the 15 cycle single-phase locomotives on the Pennsylvania Railway, and compares it with one of the 15 cycle three-phase locomotives used on the electrified portions of the Italian State Railways. Both these locomotives have been described in this column, but the comparison which Mr. Valatin draws is distinctly favourable to the three-phase locomotive. In considering the relative performances of these locomotives it should be noted that the torque of a three-phase motor is not fluctuating, like a single-phase motor, hence the former has a considerable advantage which is inherent to it.

TABLE I.

Comparison of Single-Phase and Polyphase Electric Locomotives.

	Single-Phase.	Three-Phase.
Total weight of locomotive	125 tons.	59 tons
Adhesive weight	90 ,,	41·2 ,,
Weight of motors	33·2 ,,	19·65 ,,
Rated capacity of motors ...	2,000 horse power	2,200 horse power
Tractive effort	14,700 lbs.	16,170 lbs.
Length of locomotive ...	62 feet.	41 feet.
Speed at rated capacity ...	50 miles per hour.	50 miles per hour

The cost of the above locomotives may be considered as varying very nearly in proportion to their total weight. The hauling power of the single-phase motor is less, and still more important, the paying train load is also less for single-phase, as the locomotive weight must be deducted, and this is twice as great in the case of single-phase. This feature increases in importance with increase in the gradient. If the grade is 1 in 100, and assuming the motors to be loaded to their maximum capacity, the American single-phase locomotive will haul 240 tons, and the Italian three-phase locomotive 342 tons, plus their own weight. With a grade of 1 in 50 the weights hauled

are 110 and 190 tons respectively, all figured on a basis of fifty miles per hour, and a train resistance of 16 lbs. per ton on the level.

Electrically Operated Rolling Mills.

Ever since Mr. D. Selby Bigge's paper to the Iron and Steel Institute last year, a good deal of attention has been given to this subject. The apparatus employed to take the violent fluctuations of the mill load is identical with that used for electric winding engines, described and illustrated in our issue of March, 1907, vol. xvi. p. 202. The *Electric Club Journal*, of Pittsburg, gives a lengthy description of the electrical features of such an installation, and points out that an ingot mill, rolling out to billets for making rails, girders, &c., may take 9,000 h.p. momentarily. These fluctuations, 9,000 h.p. power to zero in a few seconds, would altogether upset the pressure regulation of any electric power plant, however large, were it not for the intervention of the flywheel converter which has, in this particular case, two 13 ft. flywheels, each weighing 26 tons and running at 375 r.p.m., which falls to 310 r.p.m. as the load comes on. This corresponds to a normal angular velocity of 15,400 ft. per min. for the flywheel. The part of the kinetic energy stored in the wheel, which is given up during the drop in speed from 375 to 310 revolutions, amounts to about 7,000 h.p.

It is possible with electrically driven mills, as has been shown by Mr. D. Selby Bigge, to considerably reduce the cost per ton for producing rails, girders, &c., and, if power is generated from blast furnace gas engines, the cost of power should not exceed ·25 pence per unit on a good load factor.

As it has been found that, with electrically driven rolls, the units used per ton rolled is about twenty, it follows that the product of the mill should cost about 5*d.* per ton.

FIG. 2.—PETROL-ELECTRIC OMNIBUS ON LONDON STREETS.

Some Recent Petrol-Electric Vehicles.

The omnibus illustrated in Fig. 2 is now familiar to those who travel from North to South London, *via* Westminster Bridge. It represents one more step in the evolution of a satisfactory passenger road vehicle.

It is claimed that, in addition to all the advantages of electric propulsion, viz., ease of control, even acceleration, and absence of vibration, it reduces very substantially the running cost, a feature which is of greater importance than all the others. It did not, however, receive adequate attention in the days when the motor bus fever was as its height, but which has, since then, very effectively demonstrated its importance.

Fig. 3 shows the chassis, and it may be said that the equipment consists of a petrol

engine under the bonnet, direct-coupled to a dynamo under the driver's footboard, and a motor worm geared to each of the two rear wheels, which are, by the way, mounted on a "dead" axle.

The engine is rated at 30 b.h.p. and can be run up to 1,050 r.p.m. The dynamo, which is a 4 pole machine with commutating poles, is shown in section Fig. 4, which also illustrates the method of suspension; it will give 200 amperes at 180 volts, and runs sparkless under all loads.

The motors, as will be seen from Fig. 3, are mounted on the outside of the frame, and drive the rear wheels through a short cardan shaft, and a worm gear having a ratio of reduction of 12 to 1.

Just behind the motor will be seen the brake, the drum of which is water cooled. Each cardan shaft has a brake, and close

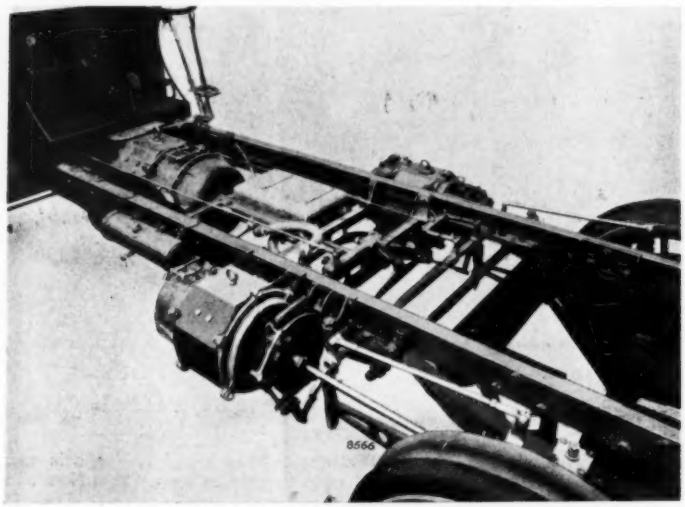


FIG 3.—CHASSIS OF PETROL ELECTRIC OMNIBUS

by the motor is a telescopic universal joint, while just outside the worm gear case is another universal joint. Connection between the worm wheel and the road wheel is effected by means of a dog clutch, so that the rear wheels may be readily removed for repairs, or the renewal of a tyre.

The engine is controlled by a throttle lever located on the steering wheel. The motor control is by a side lever which places the motors in series or parallel, and reverses. The generator voltage is controlled by a small lever on the steering wheel; this gives a fairly wide speed range, the speed being proportional to the voltage at the motor terminals. No resistances are used in series with the motors, so that the side lever gives two speeds, motors in series, and motors in parallel, and one reverse, motors in series only. All other speed changes are effected by throttling the engine, or varying the generator field, both of which are effected on the steering wheel. The equipment is from the works of Messrs. J. & E. Hall, of Dartford.

A Development in Electromobiles.

Electric vehicles propelled by the much talked of Edison Nickel-Iron Cell have at last made their appearance. It seems that all the initial difficulties which we have at one time and another described have been overcome, not by Americans, but by Germans, and a company has been formed to sell the battery and electromobiles propelled by it, in this country. Manufacturing operations will also be commenced in this country as soon as the requirements of the market have been gauged.

The nickel-iron cell has been in use on a large number of electromobiles in Berlin with very satisfactory results. Up to the present, batteries have run 40,000 to 50,000 miles, costing nothing for repairs or renewals. As it is indestructible, and cannot be damaged by over-charging, or over-discharging, can be charged in three instead of eight hours or more with a lead cell, weighs much less, and is free from the chemical troubles which always threaten the life of a lead accumulator—it may even

be washed out with the garden hose—its commercial prospects seem bright. Results in Berlin seem to indicate that the running cost of such vehicles is about one half the cost of petrol-driven automobiles.

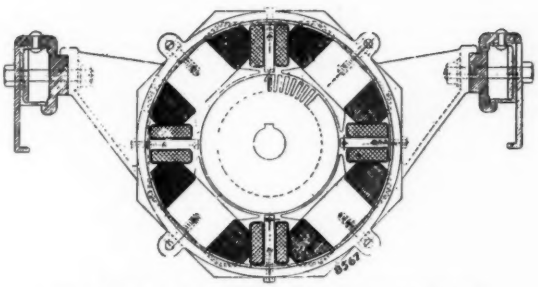


FIG. 4.—SECTION OF GENERATOR, PETROL-ELECTRIC OMNIBUS.

An Electrically Propelled Fire Escape for Liverpool.

Those of our readers who are familiar with fire brigade practice in Europe know that the use of mechanically propelled extinguishing and rescue appliances is already very extensive, and growing rapidly. Berlin, Hanover, Offenbach, Leipsic, Paris and Vienna, all have numbers of these, and electric propulsion is especially favoured in many cases. Vienna, where there are twenty-one mechanically-propelled fire vehicles has nineteen, or over 90 per cent. propelled by electricity, using storage batteries.

Glasgow has a petrol-electric fire escape, but Liverpool appears to be about the first to use an electrically driven fire escape using storage batteries carried on the vehicle. Fig. 5 shows this escape, which has so far proved absolutely reliable, and much more rapidly manœuvred than any other type. A feature of the machine is the adoption of a front, instead of a rear, wheel drive. This has been characteristic of many modern electric vehicles, and it is desirable to point out that while tests indicate a small saving in power when the vehicle is driven by motors on the front wheels, the chief value, and this is of particular importance in fire brigade work, is the great reduction in the tendency to side-slip.

When turning corners quickly, the front wheels draw the machine and the tendency of the rear wheels is to follow in the same track, but if the vehicle is pushed from the rear wheels, the tendency is for the back or pushing wheels to propel the vehicle in a straight line, consequently the rear part swings round. This reason, more than any other, has led to the rapid growth of front wheel driving with electric vehicles. The escape illustrated is in four sections, and has its trunnions carried in a steel frame fixed to the turntable. As soon as the ladders are released, two men, who hang on to the bottom, swing it into a vertical position, and a small electric motor extends the ladder to its full height of 87 ft. in 20 secs.

It is stable enough to carry a man with two lengths of hose, when unsupported by anything except its own base, and may thus be used as a water-tower. The escape has of course all the usual features of such appliances, the new features being the adoption of electric propulsion, and particularly the shortening and extension of the ladder by electric power, which, of course, vastly increases its value under circumstances when every second is of importance.

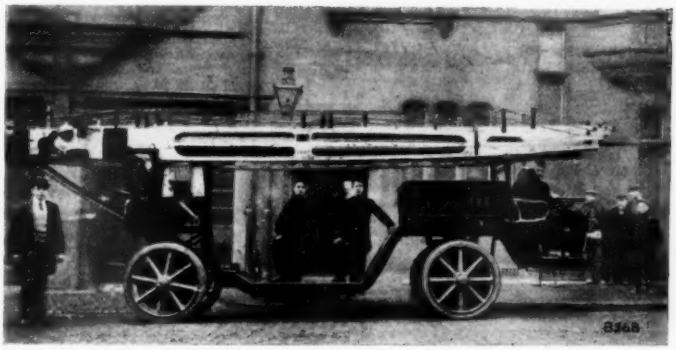


FIG. 5.—ELECTRICALLY-PROPELLED FIRE ESCAPE FOR LIVERPOOL FIRE BRIGADE.

Iron, Steel and the Foundry.

Conducted by PERCY LONGMUIR.

BLAST FURNACE PRACTICE.

Mr. J. J. Porter* in a lecture on "Recent Progress and Present Problems," gives a comprehensive survey of current American practice and future developments. From a technical point, blast furnace progress may be classified under: (1) Increased production, (2) fuel economy and (3) labour economy. In 1887 the American pig-iron output was about 6,000,000 tons; to-day an output at the rate of 27,000,000 tons per year is obtained. In the same time the number of active furnaces has increased from about 350 to 450, and from this it is evident that the average capacity of individual furnaces has been more than tripled in twenty years. In 1887 the most favoured sizes were a height of 70 ft. by 15 ft. at the greatest diameter, with a volume of about 11,000 cub. ft. and an average output of about 100 tons per day. When the movement towards large furnaces had reached its culmination, the average new furnace was about 95 ft. by 22 ft., with a volume of about 22,000 cub. ft. and an output approaching 400 tons per day. During this period the furnaces have just about doubled, and the further increased production is due to improved metallurgical methods resulting in more rapid driving. One aspect coming under the last head is that of the large use of Lake Superior ores, first introduced in a small way in 1854. In 1887 these ores furnished 50 per cent. of the pig produced, and to-day nearly 80 per cent. of the total product is made from them. The increase due to this cause would have been greater but for the realisation of the fact that the "inexhaustible" cannot apply even to the most extensive deposits. The fuel ratio is dependent chiefly on the chemical composition and physical structure of the stock used, but is also subject to a certain extent to the skill of the furnace manager.

* Cincinnati section of American Chemical Society.

The effect of larger furnaces on output has been noted, but at the present time the tendency is towards a somewhat smaller furnace and a slightly lower average output, the reasons being purely metallurgical and connected with the fuel consumption of the larger furnaces. Favoured sizes at present are, for furnaces making low silicon iron from Lake Superior ores, 85 to 90 ft. by 22 ft., and for furnaces making foundry iron from the same ores about 80 by 20 ft. Unsatisfactory results from the largest furnaces are chiefly due to the inability of the coke to meet the heavy pressure and the difficulty of getting a proper blast distribution. Another feature lies in the necessity of even distribution of the stock at the top; the net result of these and other factors is to limit, under present conditions, the most favourable height of furnace to from 80 to 90 ft., with a bosh diameter of about one-fourth the height.

On the whole the furnaces of to-day do not equal the low records for fuel consumption made in the preceding decade. Probably the best record is that of No. 6 furnace of the Illinois Steel Co., which during 1897 averaged 1,580 lbs. of coke per ton of iron. During 1890 to 1900 average fuel consumptions of 1,800 to 1,900 lbs. of coke per ton of iron were common for furnaces using Lake Superior ores, whilst the same furnaces to-day are using 2,000 to 2,200 lbs. The cause for this poor thermal efficiency has been earnestly sought, and after years of study the weight of opinion seems to incline to the belief that it is due to mistakes in furnace construction, these in turn being due to a too great reliance on engineering and a corresponding neglect of metallurgical principles. A classification of these mistakes results in too large furnaces, wrong lines of furnaces, and mechanical filling devices which cause an unequal distribution of stock. These items are followed out in full detail and with special reference to the

fine Mesabi ores. Taking one example, the fine ore being reduced high up in the furnace and at the same time swelling in volume is pressed against the sides forming an arch of compacted material which offers resistance to the passage of blast. A space forms below the arch, and when the scaffold does give way the large volume of high pressure gas beneath produces, by its sudden release, an explosive effect. Evidently then, the greater the slope or overhang of the inwall, the more space for the expansion of the reduced ore, and the less chance to form arches. Recognition of this in furnace construction has resulted in the use of from 80 to 100 per cent. Mesabi ore, with less serious slipping and lower fuel consumption than is customary in furnaces using less than 50 per cent. In regard to mechanical fi lers, early types were troublesome owing to unequal distribution, a defect largely remedied by rotary distributors.

In discussing problems Mr. Porter confines himself chiefly to the south as typified in the Alabama district, "that country where Nature has done so much and man so little." The percentage distribution of iron making costs at northern and southern furnaces is somewhat as follows :—

	Northern.	Southern.
Ore ..	50 ..	30
Flux ..	2 ..	2
Fuel ..	32 ..	50
Labour ..	8 ..	10
Maintenance ..	8 ..	8

These figures show the opportunities of fuel economy in the case of southern furnaces. Promising lines for work in this direction are in the first place a smaller volume of slag, and as this is regulated by the content of silica and alumina in the charge, the problem is resolved into mining and concentrating methods in the case of the ore and into washing and coking methods in the case of the coke. The effect of dry air blast on fuel economy is recognised, and it is well to note that other methods of drying than refrigeration are possible and may prove attractive problems. The production of low silicon iron is also an important factor in fuel economy. The largest saving in the line of gas economy is possible through the use of the gas engine. The problems of the large unit gas engine are many and offer an attractive field for work. Another direction lies in decreasing the weight of gases, which is theoretically possible by concentrating the oxygen in the blast, or in other words eliminating part of the nitrogen. To-day this is merely a dream of the future, but it is an interesting possibility. Unless the electrical furnace displaces the blast furnace, we may someday witness enormous combination plants, where iron is made with concentrated oxygen blast, whilst the residual nitrogen is converted into fertiliser.

STEEL: ITS PRODUCTION AND PROPERTIES.

The "Chute" Steel-making Process.—This process evolved by Mr. H. O. Chute is described in a recent issue of the *Iron Trade Review*. Reference to the accompanying diagram will show the leading features; thus in the blast furnace *A* phosphoric and siliceous ores are smelted with a high fuel ratio in order to obtain a superheated iron of high silicon content thereby passing most of the sulphur into the slag. On tapping the furnace the iron flows by gravity along the runner *B*, which may be lined with iron ore or mill scale. Slag and kish are skimmed off at *D* with the skimmer *C*. During the passage of the iron along the runner *B* fresh additions of ore are made from the container *E*. The action of the ore lining the runner, and that added to the surface of the flowing metal results in some oxidation of the silicon and manganese, and the cinder formed is removed at *G* by skimmer *F*. On reaching the vessel *H* the metal is said to be in ideal condition for treatment in the basic open hearth. This vessel serves as a mixer in which the iron may be held until it is transferred to the open hearth furnace *K*, and there worked down into finished steel.

Protective Coatings for Iron and Steel.—In a paper read before the Students' Engineering and Metallurgical Society of the Sheffield University, Mr. V. C. Faulkner reviews in full the various processes followed for the protection of iron and steel. The methods described include the more usual protective processes, galvanising, tinning, painting, electro-deposition and Sheradising. In discussing the disadvantages of hot galvanising, attention is drawn to the deterioration in strength, a feature specially marked in the case of thin sheet and wire. The delicate temper of hardened steel wire is also destroyed. Cold galvanising, or the deposition of zinc from an aqueous solution of one of its salts, is conducted with a view to avoiding this loss of strength. The difficulties of the earlier methods have been largely overcome in the "Cowper Coles" process. In this method the solution of zinc sulphate is continually pumped into regenerating tanks and returned to the galvanising tanks by gravity. The regenerating tanks contain zinc dust and the

solution is maintained at a temperature of 21° C. The coating obtained adheres more firmly than that obtained by hot galvanising, and the articles are not weakened by immersion in the bath.

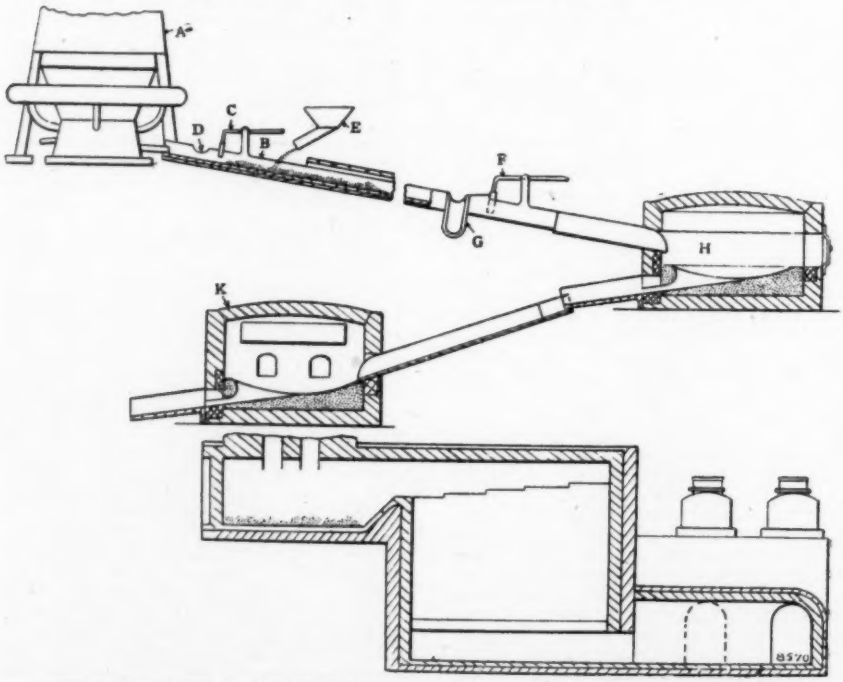
Occurrences and Uses of Vanadium. — The opening up of a large sulphide deposit in Peru has, according to Mr. J. Kent Smith,* considerably brightened the outlook for vanadium. An analysis of this ore shows 39·84 per cent. vanadium sulphide, and being free burning the ore is calcined with ease, losing 45 per cent. of its weight in the process. An analysis of the calcined ore shows 58·08 per cent. vanadic oxide. Previously the chief commercial sources of vanadium have been the Spanish lead ores. As mined, these contain only 4 or 5 per cent., but by hand dressing could be concentrated to about 12 per cent. vanadic oxide. These figures show the value of the sulphide deposit. Further, the comparative ease with which vanadium can be extracted, as opposed to the separation of smaller quantities from complex ores, make it a great commercial consideration.

* The *Mining World*.

FOUNDRY PRACTICE.

Converter, or Small Open Hearth.* —In his third instalment of this discussion Mr. W. M. Carr reviews some of the requirements of the converter process, laying special emphasis on the high grade of pig-iron required. After discussing the generalities of open hearth practice, the author passes to a consideration of refining efficiency. As a melting unit, the open hearth plant is slow, but as a refining unit it cannot be surpassed. This is shown in the fact that after melting samples may be taken and tested in order to definitely ascertain the process of the bath. The information so gained decides the further conduct of the bath, and hence the personal equation is reduced to a minimum. In the crucible process an error in the initial charge is

* The *Iron Trade Review*.



GRAPHICAL ARRANGEMENT OF BLAST FURNACE AND OPEN HEARTH PLANT, PROVIDED WITH A MIXER SHOWING APPLICATION OF CHUTE PROCESS.

carried through the entire operation. With converter practice it is not possible to make corrections in the cupola after charging, nor is it practical to stop the blowing operation in the converter in order to sample for chemical tests and thereby forestall possible errors. An open hearth furnace has the advantage of long runs, and may be built in any capacity from 2 to 2½ tons. A five-ton furnace is capable of turning out two heats per ordinary working day, and with no greater labour than that required to operate a single 2-ton converter. With cheaper melting stock for the open hearth, the cost of production will not be heavier, even with extra fuel to keep the furnace hot over night. This point requires confirmatory data, which Mr. Carr promises in later articles.

The Manufacture of Large Castings.—In a paper read before the American Society of Mechanical Engineers, Mr. W. A. Bole discusses the relation between the foundry department and the department of engineering design. The subject is extensive, but towards the end of his paper Mr. Bole gives some very pertinent advice. It might seem almost unnecessary to say that shapes of sections, flanges, and other projections should be so designed that the moulder may most easily produce them without having to resort to complicated methods. A designer with practical experience in foundry work will see many opportunities for making shapes that will "draw" easily, rather than other shapes which may look as well on paper but are much harder to produce. Of course the founder can produce almost any conceivable shape if he has to, but engineering design is at its best when the shapes are at once suitable for the intended purpose and easily and cheaply produced. The designer should try to put himself in the moulder's place and imagine himself making the mould in question. Then he will see that a small difference in design sometimes causes a big difference in cost and risk.

Workshop Practice

Cutting a Multiple lead quick pitch thread on the Milling Machine.—Fig. 5 shows an ignition control screw, for an automobile, ¾ in. diameter, and 4 in. long, having a thread cut on 6 in. pitch, with eight starts. This is an extremely difficult and costly operation, even if done on an universal milling machine, whilst to cut the thread in a lathe is out of the question both as regards speed and accuracy. Having a quantity of these to make, we designed the fixture shown for that purpose. It is used on a small hand-feed milling machine, and proved up to expectation for both speed and accuracy. With it a boy can average four screws per hour, or forty screws per day of ten hours, which cannot be improved on by any other method I have seen, even though costing more money.

A is a ribbed cast iron bracket, with two round bosses on the top. The front boss is bored out 2 in. diameter to allow the work spindle, *C*, to be a nice sliding fit: the back boss is threaded to suit the sleeve *B*, on which a master thread is accurately cut, the same pitch as the work intended to be operated upon, 6 in. in this case. Fig. 1 shows the elevation and Fig. 2 the end view. The work spindle *C* is shown in Fig. 3. *D* is a handwheel, 9 in. diameter, keyed fast on the sleeve *B*, *E* is a steel lever screwed on to the end of *C*, secured by the locknut *G*. On the outer end of *E* is a spring plunger *F*, which acts as the dividing mechanism for the eight threads, by operating in the hard steel bushes, *H*, in the rim of the handwheel *D*. *I* is an ordinary spring collet, for gripping the work whilst being operated

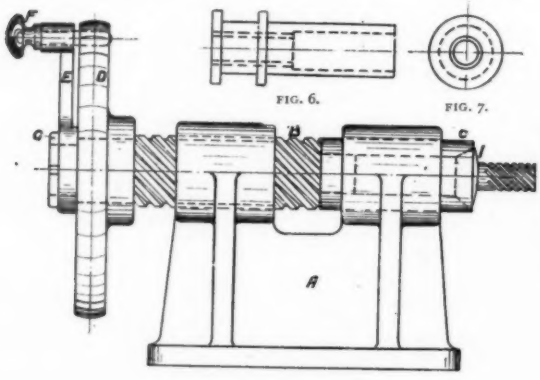


FIG. 1.

FIG. 6.

FIG. 7.

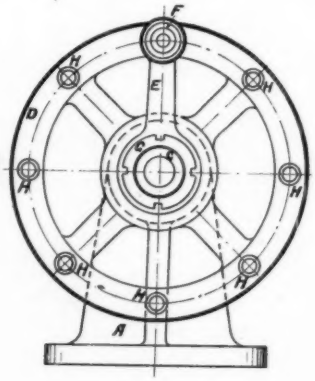


FIG. 2.

FIG. 3.

FIG. 4.

FIG. 5.

CUTTING A MULTIPLE LEAD QUICK PITCH THREAD.

upon; it is tapped out ½ in. diameter at the back end to receive a long bolt which passes through the hollow spindle *C*, and is drawn tight into position by a nut, at the back of *G*, not shown. The work is gripped in the fixture as shown in Fig. 1, and the fixture is set at an angle to the cutter spindle and clamped down on the machine table. This angle varies according to the pitch of the thread to be cut, and is found by experimenting. The cutter is 1¼ in. diameter, and made to an angle of 29 degs., the same as a standard worm cutter. It has a shank ½ in. diameter, turned solid with it; this is gripped in the milling machine spindle in exactly the same manner as the work is gripped in the fixture as shown in Fig. 1. The table is raised to the correct height, and the operator commences to turn the handwheel *D*, which feeds the work under the cutter at the same speed as the lead of the master screw *B*, therefore a thread or spiral, of the same lead, is reproduced in the work being operated on. After one thread is cut, the work is brought back by reversing the handwheel, and the dividing takes place, ready to commence the next thread. This is accurately accomplished by the ingenious arrangement shown. On pulling the spring plunger, *F*, out of the handwheel *D*, the work can be revolved quite free and independent of the master screw *B*. The plunger is dropped into the next hole in *D*, and the work proceeded with. It will thus be seen that it is quite impossible to divide the work wrongly, and should the operator miss one thread, he can turn the wheel back and commence again well knowing that a mistake in the dividing cannot possibly occur as long as the spring plunger *F* is in the bushes in the handwheel *D*. Of course the number of bushes in *D* must agree with the number of threads to be cut on the work, *e.g.* eight holes for a screw with eight threads on. The same wheel can also be used for cutting four and two threads, but for six or ten threads other dividing wheels would have to be substituted.

I may add that these screws now cost us about one penny each for labour.

The brass nut for this screw to operate in is shown in Figs. 6 and 7, and is tapped out at one end, as shown by the double dotted lines. This also presented some difficulty, as owing to the quick pitch of the thread, or spiral, and the tap being so small in diameter, viz., ⅜ in., we could not feed it through as an ordinary tap. We solved the problem successfully by gripping the tap in the fixture in place of the screws, and held the nuts in a fixture central with the tap, and by simply turning the handwheel, *D*, round, the tap was fed through the nut, and made a good thread of accurate pitch.

This fixture is comparatively cheap to make taking into consideration the nature of the work involved, and the high standard of accuracy required in the work. It has saved its cost many times over, as the only other means of doing this work satisfactorily, that we could think of, was either by a universal milling machine, or a thread milling machine, and either would cost a very considerable sum of money, and would then not be as cheap to operate as the fixture here shown. S. L.

An Emergency Rig-up in the Smith's Shop.—In the year 1902, when I had charge of an engineering works in West Cumberland, we had an offer of a contract for a large number of ties or cramps for stone work connected with a sea breakwater. The cramps were to be made double ended, as shown in Fig. 1, from 3 in. square mild steel, and the section at the ends had to be fully maintained. Our

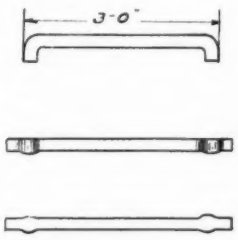


FIG. 1.

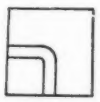


FIG. 7.

FIG. 3.

AN EMERGENCY RIG-UP IN THE SMITH'S SHOP.

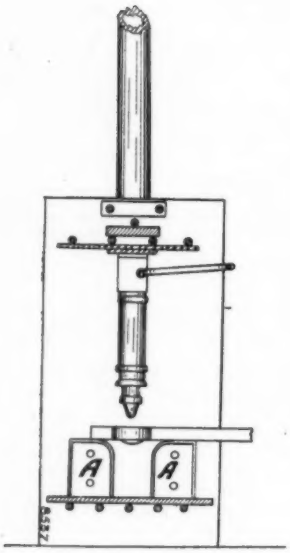


FIG. 5.

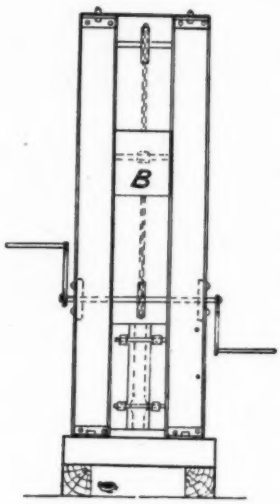


FIG. 2.

AN EMERGENCY RIG-UP IN THE SMITH'S SHOP.

equipment in the smith's shop was of an order suitable for light work only, so it was out of the question for larger section bars to be employed or to forge them with ends formed out of the solid. The principals of the firm were, however, anxious to carry out the contract, so they gave me a free hand to adopt what methods I thought fit to carry out the work, providing it did not involve any considerable expense. I should mention that the cramps had to pass rigid inspection after completion.

The method adopted was the following: All the material from the mill was ordered sawn to an exact length sufficient to make a complete cramp, having ascertained beforehand the length necessary for each bend.

A heavy perforated slab (of which we had a number lying about) was placed flat on two wood blocks, and to it were fixed two 8 in. by 4 in. upright girders 10 ft. in length, braced at the top so as to leave a space of 8 in. between, as in Fig. 2. A cast iron block, Fig. 3, was then made, with an impression sunk to take a 3 in. square bar, this being hollowed out and recessed to form a swell at the part where the bend was to be. Cast in the block were four staples which received two cotters to secure a loose front plate which was placed in position after the 3 in. bar was set upright in the block. The block was fixed on the slab between the two uprights. A loose block or monkey was also arranged to slide between the upper length of the girders at B, Fig. 2, and at the lower and back side of uprights was fixed a small shaft with two winch handles attached to an 8 in. chain wheel. A similar wheel was also fixed at the top of the uprights. Round these wheels travelled an endless iron chain $\frac{5}{16}$ in. diameter. Attached to the monkey was a simple catch operated by a lever for clipping the chain, which in travelling upwards would thus take the monkey up with it, the latter being released and allowed to fall after reaching a certain height. The release took place by the action of a trigger when brought into contact with a lever on the monkey.

The heats were taken from ordinary pot fires built on the hearths, but so arranged that no heat was more than 3 in. in length. As soon as the bar was hot it was placed on end in the block, the front plate was cottered in position, the smith's mates turned the handles, thus lifting the monkey, which was allowed to fall on the end of bar on being released as above described. Three blows were found sufficient to upset the bar and fill up the recess in the block, the bar being left the correct size for bending without any loss of area below 9 sq. in.

The bending of the cramps necessitated considerable scheming and forethought.

A perforated slab 8 ft. by 3 ft. by 8 in. was placed on end and bolted to an upright column of the shop as in Fig. 5. On this slab was secured by clips an ordinary 5-ton hydraulic lifting-jack. At the top of the jack to take the thrust was placed a bar with a number of $1\frac{3}{4}$ in. square pegs inserted in the slab. On the lower part of the slab was secured two short ends of 12 in. by 4 in. girders shaped as shown at A, under which were placed several $1\frac{3}{4}$ in. square pegs, to receive the thrust. A U-shaped

piece of steel made to correspond with the inside radius of the bends was screwed into the foot of the jack in a hole previously drilled and tapped.

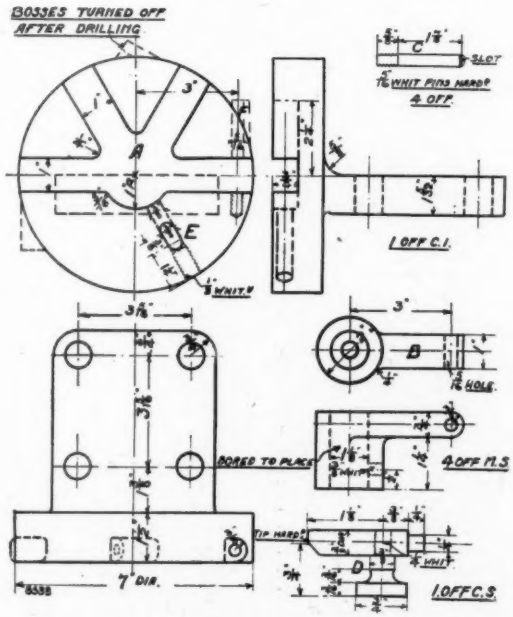
Everything being ready the same length heat was taken on the swelled parts of the bars, and when hot they were placed across the ends of girders (A, Fig. 5). The smith's mate then worked the jack causing the foot to descend and so form the bend. In the bending process the bars pulled to the right size at the back, but thickened a little on the inside, so with the same heat we placed them in a cast iron block shaped as shown in Fig. 7, and with two blows of a light steam hammer were brought to the correct shape and required no more doing to them. Each corner was made in two heats by one smith and two mates. The contract turned out a profitable undertaking, and in no instance was one rejected. I should mention that both ends of the bars were upset before bending.

F. W. B.

Tool Holder for an Ordinary Lathe.—The following arrangement was devised to enable four operations such as drilling, facing, boring, and tapping, to be carried out on one machine, and by its adoption an ordinary lathe may be practically converted into a turret-headed lathe when one of the latter is not available, and will produce the same results. The device is not limited to the above operations, but, as may readily be seen, can be arranged differently to suit any job. In the first place a casting is required as shown at *A*. This has four slots arranged to carry four chucks, *B*, which work off a hinge on the outside edge of the front plate *F*. The small pin *C* acts as fulcrum for the chucks to work on. In order to ensure the chuck being flat down, a small bayonet-catch is inserted on the under side as indicated on the sketch at *E* and the bolt *D*. When tools are out of use they are thrown back out of the way and will not interfere with the working of the lathe. The apparatus is placed on four pins of the tool box and secured by the nuts used for ordinary purposes. The initial outlay is insignificant, and the apparatus will, if readers copy, amply repay them for their trouble.

I have given this a practical test myself and am very well satisfied with its results.

C. J. F.



TOOL HOLDER FOR AN 8 IN. LANG LATHE.

Machinery Appliances and Processes.

The Editor invites advance particulars, drawings, blue prints, photographs, and any data that will enable him to publish an adequate description in these columns of any new work, machinery, or installation, possessing special features of interest. The cost of publication and engraving is borne by us, and no compensation is expected. In the endeavour to produce a really high-class and technically valuable machinery section the Editor seeks the co-operation of engineers and manufacturers.

Oil Motor for Agricultural Purposes.

It has long been recognised that the oil engine, by reason of its simplicity, ease of manipulation and freedom from boiler troubles, possesses a peculiar value above all other types of prime mover, for use in all out-of-the-way places. A further step was needed, however, to render it of utility in agricultural work, and that was the production of a design which would include the property of locomotion, hence enable it to perform the duties of hauling, ploughing, and cultivating, ordinarily undertaken by horses or by traction engine and steam plough.

Messrs. Marshall, Sons & Co., Ltd., of Gainsborough, were among the first to recognise this, and have occupied themselves for some time past with the development of an oil motor suitable for use in such farm duties as those above mentioned. In the result they have perfected an engine of a type shown in the accompanying illustrations which we are assured is a thorough success.

For a year past one of these motors has been at work in the Gainsborough district undergoing tests in ploughing, cultivating, thrashing, and hauling, culminating in a 24 hours' non-stop ploughing run at Grayingham, Lincs. During the 24 hours the engine ran continuously without a stop, hauling a six-furrow plough. The results were that 22 acres of land were ploughed on a total fuel consumption of 44 gallons of paraffin, and, moreover, without any addition to the water supply carried by the engine for cooling purposes.

A feat of this character is one that is particularly noteworthy, when it is remembered that the slow speed attained in ploughing permits of little or no cooling action of the air, as in the case of a motor omnibus or pleasure car, and that the water cooling and circulating apparatus must therefore be particularly efficient.

The chief features of the motor are as follows:—The engine is supported on a channel steel frame, the water cooler being carried at the front end, and an extra water tank at the back end. The engine is a two-cylinder one, with the valves arranged on top of the cylinders, and worked by means of bell-crank levers from the cam shaft. This cam shaft in turn is driven by spiral gearing, and a vertical shaft from the engine crankshaft. The two cylinders, which are 6½ in. diameter by 7 in. stroke, are cast in one piece with the water jacket. At its normal speed, viz., 850 r.p.m., the engine develops 30—35 b.h.p.

The carburetter is of the firm's own manufacture, and is the result of several years experimenting. It is exceedingly simple, with no parts likely to get out of order, and is suitable for running with petrol, benzine or paraffin, without any alteration. It consists of little more than a corrugated plate forming the top side of a flat box. Below the plate pass the exhaust gases; above it the air and petrol mixture.

The exhaust valves are surrounded by a water jacket, and the circulating pump is of the gear type. The governors are of the ordinary bell pendulum type, and act upon the throttle valve. The flywheel is fitted with an efficient clutch suitably designed to transmit the full power of the engine.

The motor is fitted with three speeds for 2, 4 and 6 miles per hour, with reverse on the slowest speed, and is able to perform the maximum amount of ploughing, or to ascend hills with a load, and to minimise time by running quickly on the roads with light loads. A foot brake is fitted on the

OIL MOTOR FOR AGRICULTURAL PURPOSES.

The above illustrations show—(1) The motor hauling a 4-furrow plough. (2) Driving a 4 ft. 6 in. finishing-thrashing machine. (3) Hauling three loaded farm wagons

second intermediate shaft, and a hand brake acting upon both hind wheels is also supplied. The hind wheels, which are the driving wheels, are built up. The rim is of steel and the spokes are of round best quality iron: the bosses are of steel suitably lined with brass bushes.

The radiator or cooler is of an entirely new design, and, as stated above, is very effective. Provision is made for replacing the slight amount of water lost by evaporation and possible leakages, which need not exceed more than two to three gallons per day.

The water from the jackets, having been pumped to the top of the cooler, is allowed to fall in a spray, through a series of oppositely placed corrugated baffle plates, to the base of the cooler.

As the water descends it is met with an upward air draught induced by the exhaust from the silencer which is fitted on top of the radiator. It should be mentioned that no fan is required in the cooling.

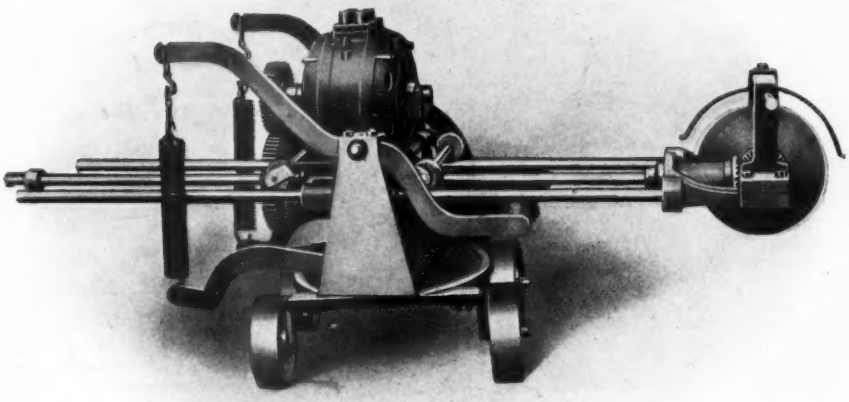
The engine is started with petrol, and after running for a few moments the petrol cock is closed and the paraffin cock is opened, the supply tank for paraffin being kept filled to a constant level by means of a small pump worked by an eccentric from the cam shaft. The fuel supply carried is sufficient for two or three days' work. The driving pulley is fitted on the end of the crankshaft, suitably placed for driving any kind of machinery by means of a belt, and the motor can be readily set in position in line facing the machinery to be driven. The weight of the motor complete in working order is $4\frac{1}{4}$ tons, and as this weight is distributed over four travelling wheels, very little damage is done to the roads or to headlands with ploughing.

The powers and usefulness of the engine are sufficiently well indicated by the illustrations.

Portable Electric Grinder.

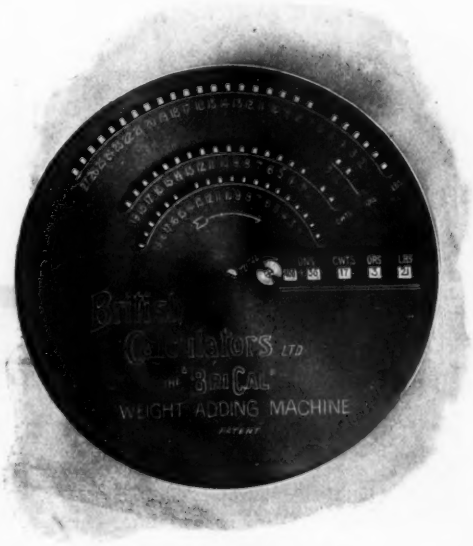
The accompanying illustration is that of a new type of portable electric grinder designed as a substitute for the flexible shaft type hitherto extensively used for grinding the face and grooves of tramway rails, uneven surfaces on castings, and

OIL MOTOR FOR AGRICULTURAL PURPOSES.



PORTABLE ELECTRIC GRINDER.

generally in shops where any grinding has to be done. It is claimed that the new machine possesses the important advantage that it is practically free from vibration and that one man suffices to attend to it. As will be seen the grinder is entirely self-contained, the motor, of which any suitable size or type may be used, being carried on the machine itself. The emery wheel is driven, by means of spur gearing at the rear of the motor, through a centre shaft and specially designed bevel gearing in the grinder crosshead. The two side shafts act as guides, and all three shafts have a sliding motion through a bracket at the base of the motor, thus admitting of an effective range of grinding of about 3 ft. in either a vertical, longitudinal, or radial direction. A swivelling motion can also be obtained permitting grinding to be done at any angle. Any size of emery wheel can be fitted on the spindle and a change can be effected in a few minutes. The machine may also be used as a fixed grinder for grinding tools on the road. It is being manufactured and sold by



A WEIGHT ADDING MACHINE.

Messrs. Geo. Robson & Co., Napier Street, Sheffield.

A Weight Adding Machine.

A new calculating machine, known as the Brical Weight Adding Machine, has recently been brought out by the British Calculators, Ltd., of the Invicta Works, Stoke Newington, which is operated practically in the same way as their well-known money-adding machine. It is used for the addition of weights, and deals with pounds, quarters, cwts., and tons, converting automatically quantities in one series to quantities in terms of another series without mental division. An idea of its appearance may be gathered from the illustration on page 199. It is disc-shaped and has four curved slots corresponding to the four series of weights dealt with. The white spots visible on the slots are wheel teeth, the space between each representing and being marked by a number. In the windows below the results are indicated. If, for example, it is desired to add six and five, a small rod is placed in the space opposite six and the wheel pushed round to the end of the slot, when six will appear at the window. The small rod is next inserted in the space opposite five, and on the wheel being pushed round as before the total, eleven, will be read off at the window. The conversion from pounds in quarters, and so on, is, as above explained, carried out automatically.

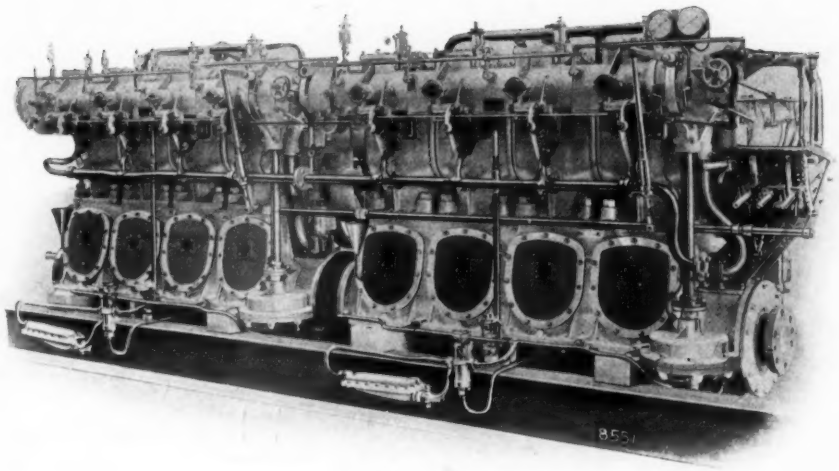
PORTABLE TWIST DRILL AND TOOL GRINDER.

Portable Twist Drill and Tool Grinder.

One of the handiest and most compact forms of portable tool grinder that has hitherto been brought to our notice is one which has recently been introduced by Messrs. S. Holmes & Co., Bradford, who have devoted a great deal of attention to the design of twist drill grinders. This, their latest machine, is intended for users who have occasion to work away from the source of power, and, as will be apparent from the illustration given, it may be carried easily from place to place, and is moreover always ready for work. To ensure rigidity, the machine, which weighs approximately 60 lbs., is mounted on a tripod stand. It is driven by a hand wheel through machine-cut steel chain wheels and roller chains, the maximum number of revolutions being obtained when the hand wheel is turned at about 80 r.p.m. It is indexed for different sized drills up to the maximum accommodated, viz., $1\frac{1}{8}$ in. diameter, and will take either taper, square, or straight shanks. The adjustments are simple; the pointer is set to the size required, the drill placed in the holder, and grinding may commence. The opposite end of the shaft carrying the twist drill holder is fitted with an adjustable tool rest. Some of the leading dimensions are as under: distance from floor to table top, 2 ft. 6 in.; distance to centre of driving wheel, 3 ft. 8 in.; size of table, 12 in. by 12 in.; size of driving wheel, 12 in.; floor space, 3 ft. 6 in. by 2 ft. 6 in.; size of grinding wheels, 6 in. by $\frac{3}{8}$ in.

Paraffin Motors for Submarines of the Italian Navy.

The accompanying illustration shows a set of paraffin motors constructed by Messrs. J. I. Thornycroft & Co., Chiswick, for two twin-screw submarine boats of the Italian Navy. They are constructed in



PARAFFIN MOTOR FOR SUBMARINE BOATS, ITALIAN NAVY.

four-cylinder units, bolted together, making eight cylinders on each shaft, the dimensions of the cylinders being 12 in. diameter by 8 in. stroke.

The engines are totally enclosed and lubrication is forced to all bearings; those on the crankshaft, and also the bottom of the crankcase, being watercooled. In addition, the pistons and crankcase generally are continuously cooled by sucking cold air which is drawn through by an exhausting fan. Facilities are provided for the removal and replacement of the pistons and connecting rods through the doors in front of the engine, thus rendering removal of the cylinders and access from the back of the motor unnecessary; an advantageous feature for submarine boats. Cooling water is circulated by a separately driven centrifugal pump, driven by either an electric motor, or a small oil motor using the same fuel as the main engine. The advantage of this method is that the motor may be cooled more rapidly after running, in case examination or adjustment become necessary. The separate drive also allows of a close regulation of the quantity of water to suit the needs of the engine, according to the nature of fuel used.

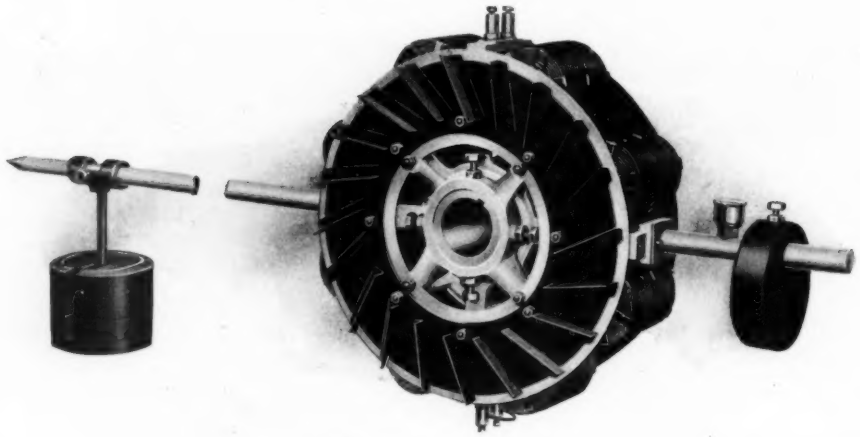
The special paraffin vapouriser is fitted with a variable exhaust by-pass so that the temperature may be exactly graduated to the needs of the particular fuel used. This is very essential as the oils used for fuel vary so greatly in the amount of heat required to vapourise them, and excess of heat is as harmful as too little. With a proper regulation of heat it is stated that the engine or vapouriser never requires to be cleaned as there is no tar deposit. Trouble on this score is claimed to be absolutely non-existent with these engines. An entirely novel feature is the reversing mechanism. Instead of the usual method of using a double set of cams for running in either direction, the camshaft is so arranged that, whichever direction the engine itself runs, the camshaft turns in one direction only, and to provide for this a reversible bevel drive, with positive clutches, is fitted. Starting in either direction is effected by means of compressed air, the valves controlling which are cut off immediately the engine is firing. Low-tension magneto ignition and distributor are fitted in conjunction with a special form of make and break gear in the cylinders.

The consumption of fuel is very low, being about ·7 pint of paraffin or petrol per b.h.p. per hour. The engines are designed with ample bearing surfaces, and are in every way suitable for heavy continuous service. The total weight per four-cylinder set is 70 cwts.

The following table of results of a three-hours' trial is of interest:—

TRIAL OF F4/1 AND 2, 8-CYLINDER 12" DIA. BY 8" STROKE, REVERSING.

Fuel Used...	...	Phoebus Paraffin S.G. ·820.
Consumption	...	·66 pint per b.h.p. per hour.



EDDY CURRENT TESTING BRAKE.

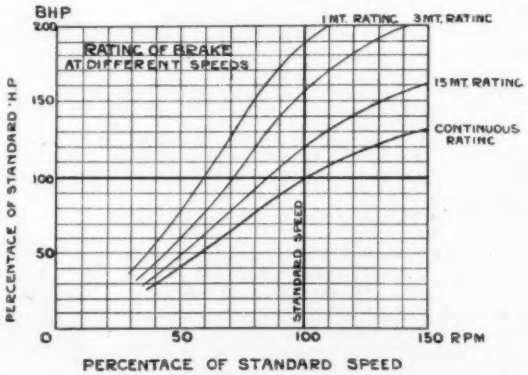
TRIAL OF F4/1 AND 2, 8-CYLINDER 12" DIA. BY 8" STROKE, REVERSING—*continued.*

Time from start. (hours).	B.H.P.	Revs.
0	324	550
½	328	555
1	340	590
1½	327	595
2	318	595
2½	318	600
3	324	600
Average revs. per minute		582·5
Average power		325·5 b.h.p.

Eddy Current Testing Brake.

Some time has now elapsed since Messrs. Frampton & Paine, Coventry, placed on the market their eddy current brake for testing motors up to 50 b.h.p. The latest type of apparatus, however, embodies some important alterations by which the mechanical construction has been improved and simplified, and the cost per b.h.p. has been reduced. This has been effected by doing away with the outer poles of the older type and substituting therefor a thin iron ring attached to each disc and fitted with cooling vanes.

The apparatus as now constructed consists of two copper discs, each mounted on a cast aluminium spider. These are made fast, one at either end of a sleeve which is keyed to the shaft of the motor under test. Riding loose on this sleeve and between the two copper discs is an aluminium casting carrying a number of electromagnets wound to have alternate polarity. These magnets consist of well ventilated coils on circular cores fitted with pole pieces. To the outside of each copper disc is secured a ring of wrought iron which revolves with the copper and at the same time forms a path for the magnetic flux. These iron rings are fitted with cooling vanes which dissipate the heat generated. When the motor is running the magnetic flux in traversing the moving copper induces eddy currents which absorb the energy of the motor in heating



EDDY CURRENT TESTING BRAKE. CURVES SHOWING RATING FOR SHORT TIME RUNS.

the discs. At the same time the flux tends to drag round the magnet system and levers. By suitably adjusting the exciting current the lever floats. The power is then given by :—

$$\text{B.H.P.} = \frac{\text{Torque in lbs.-ft.} \times \text{R.P.M.}}{5{,}250}$$

The brakes serve for short time tests on from 50 to 100 per cent. overloads at standard speeds, and will run normally at anything up to 50 per cent. of their rated speed above or below. The accompanying curve shows the rating for short time runs. The makers have further brought out a simple control-panel by which the exciting current in the brakes can be regulated very uniformly over the entire range of speed. This will be found extremely useful especially where suitable regulating gear is not available.

Adjustable Gangway at Cuxhaven.

A very great improvement in the method of landing passengers has been effected at Cuxhaven by the Hamburg American Co., by the employment of an adjustable gangway. As most people know, the ordinary gangway generally employed on piers, landing stages, &c., is often pitched at very inconvenient angles, making the landing and embarking of passengers, luggage, &c., somewhat difficult, especially if the sea is rough. With the apparatus at Cuxhaven, which has been designed and patented by Captain Iron, the Harbour Master of Dover, this is avoided, as the gangway can be adjusted to suit the height of the vessel's deck at all states of the tide; moreover it is firm and steady in all weathers, rapidly placed in position, and the landing and embarking of passengers, luggage, also horses and cattle, can be carried out much more quickly and without risk.

ADJUSTABLE GANGWAY AT CUXHAVEN.

The gangway was constructed by Messrs. Merryweather & Sons, of London, to drawings provided by Captain Iron, and consists of a rectangular angle steel tower, having a telescopic portion with a platform arranged to rise and fall, and with a range of elevation equivalent to the maximum rise and fall of the tide. The platform, on being adjusted to suit the height of the vessel's deck, is connected thereto by an ordinary platform or bridge, giving a practically level footway. The telescopic portion is raised by means of a vertical square threaded screw spindle, revolved by suitable spur wheel gearing from a winch handle at the bottom. The tower is carried on a substantial steel carriage, mounted on flanged wheels for running rapidly on rails along the quay. A substantial steel gangway, with lattice sides and a wood floor, is hinged at one end to the rising and falling platform, and at the other is mounted on flanged wheels for running on the rails. This hinged gangway is of suitable length to give easy descent from the tower platform, and, not being supported by the ship at either end, is always steady and safe, whilst the gangway from the tower platform to ship, being supported at both ends at equal height, is comparatively little affected by the motion of the ship, even in rough weather. Another advantage of the arrangement is its adaptability for use in narrow spaces, where an ordinary gangway could only be put into position at a very sharp angle. Captain Iron has had one of these adjustable gangways in use at Dover some time now, and it was after seeing this one in use that the Hamburg American Co. decided to adopt the same system at Cuxhaven, the company being very much impressed by the convenience and satisfactory working of the apparatus. The gangway has effected a great saving of time in landing and embarking

IMPROVED LATHE CARRIER.

passengers both at Cuxhaven and Dover. The illustration shows the gangway erected in the maker's yard.

An Improved Lathe Carrier.

An improved lathe carrier has recently been introduced by Messrs. Sawyer & Co., Melton, the object of which is to provide a better grip of the work than is possible with carriers of the ordinary type and to obviate the necessity of using copper packing between the set screw and the work to avoid marking the latter. These improvements have been obtained by adopting the following mode of construction. The carrier consists of three parts, an outer portion or frame, and an inner part consisting of the holder proper and the nut. The outer frame resembles a fork and consists of two frame plates cast in one piece with a solid end. In each side plate are two holes of circular section on the one side, but cut away to an angle on the other side.

The movable part consists of a screwed spindle provided with a flat plate in which is formed a pear-shaped hole similar to those in the side plates.

The plate fits between the side plates; the spindle is screwed through a hole in the solid end.

The work is passed through two of the holes of the side plates and through the hole in the movable plate, and the nut screwed up tight, thus holding the work securely between the six angular surfaces or edges, viz., the two on the inner plate held by the screw and the four on the outside plates.

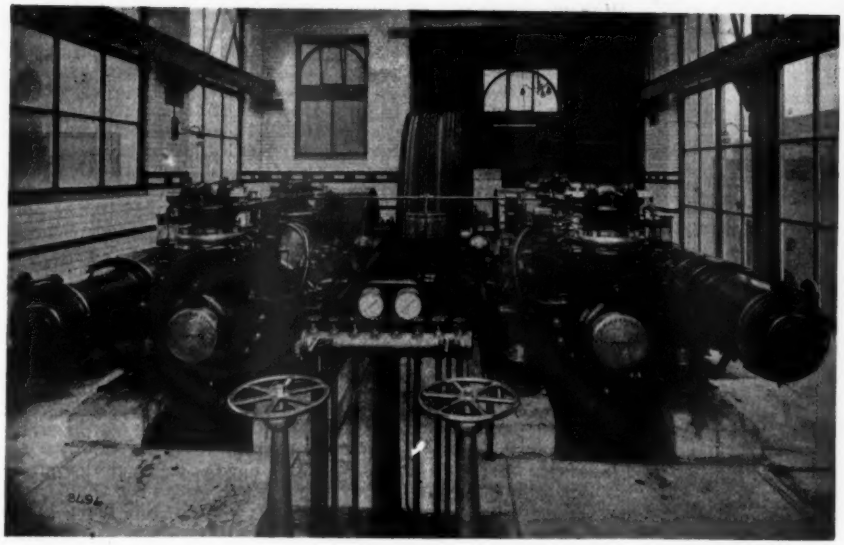
The apparatus may also be used as a grip for drilling holes through short spindles. The spindle is fixed in the carrier as shown in the illustration, the carrier being then bolted down to the drill table. In repetition work this method should save much time.

600 b.h.p. Two-cycle Gas Engine.

By the courtesy of the makers, Messrs. Mather & Platt, we are able to give some particulars and illustrations of an improved gas engine for driving part of a ring spinning mill, belonging to the Staley Mill Co. The engine, which is capable of developing 600 b.h.p at 115½ revs. per min., is of the "Körting" two-cycle type, modified, however, in many details, chiefly in connection with the charging pumps and valve gear, resulting in a simplification and reduction of the number of parts. Furthermore, the qualities of reliability, certain starting, even turning and close governing—so important in mill or electric generator-driving—have been secured, thus putting the design in possession of features belonging to the best class steam engines or turbines.

As will be seen from the photographic reproduction, the engine is arranged on the twin system, each cylinder being provided with a separate bed plate. The two engines are thus self-contained, and drive a crank-shaft much in the same way as in steam engine practice, and piston rods running through stuffing boxes to external cross-heads. The crank-shaft carries a 19-ton fly-wheel grooved for rope driving, the same shaft driving the charging pumps through overhanging cranks and connecting-rods. The detailed arrangement of the working cylinder valves and charging pumps will be understood by the aid of Figs. 1 and 2, showing sectional views of these parts.

The double-acting pump *A* (Fig. 1) for the gas is placed between the two single-acting air pumps *B* and *C*. The three pumps are thus placed in line and secured to one another as well as to the engine frame. They are, however, separated from each other by intermediate delivery valve boxes *D E*, each of which is divided by a diaphragm *F* into passages for the gas and air respectively, and is also provided with automatic valves *G H*. The gas piston *J* is hollow and is fitted on both faces with



600 B.H.P. TWO-CYCLE GAS ENGINE.

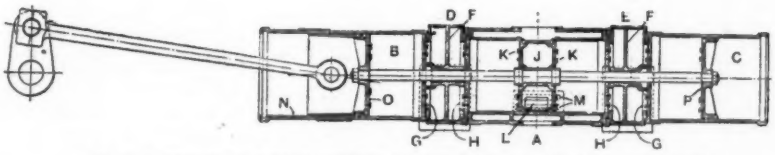


FIG 1.—LONGITUDINAL SECTION THROUGH GAS AND AIR PUMPS. 600 B.H.P. GAS ENGINE.

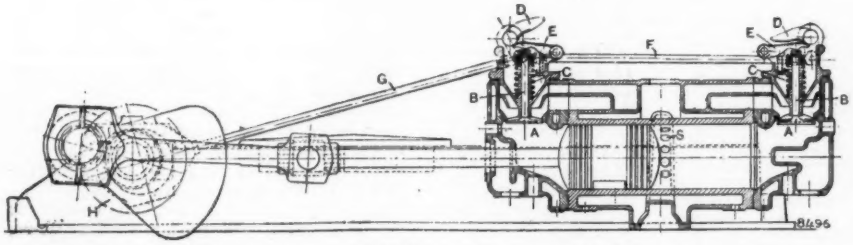
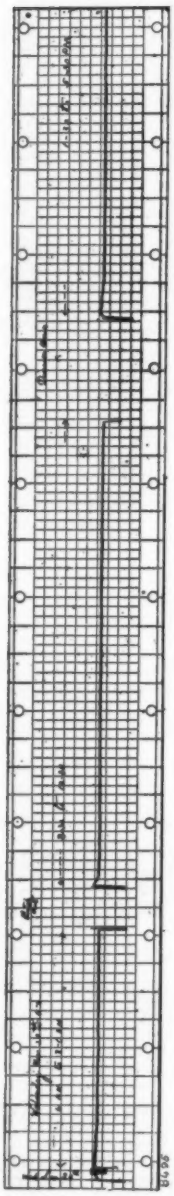


FIG. 2.—LONGITUDINAL SECTION THROUGH COMBUSTION CYLINDER. 600 B.H.P. GAS ENGINE.

automatic valves *K*. In its periphery are slots *L*. The wall of the gas cylinder is provided with suction ports *M*, which open up communication between the cylinder and gas main when the ports are uncovered by the hollow piston *J*. When these ports *M* are covered by the piston the gas is allowed to enter the piston itself, and then pass through the automatic valves *K* into the pump cylinder on each suction stroke.



MOSCROP RECORD OF A DAY'S RUN

With the object of delaying the admission of the charge of gas into the power cylinder until the latter has been thoroughly scavenged by air—a necessary process in two-cycle engines—the piston *J*, at a predetermined point in its stroke fixed according to the calorific value of the gas employed, over-runs the ports *M*, thus cutting off the gas supply, and then by its further movement forces the gas through the automatic valves *H* into one or other of the valve boxes *D E*, and thence to the gas passages to the power cylinder. The single-acting air pump *B* is provided with an extension *N*, which acts as a slide for the crosshead and connecting-rod, and by which the pumps are operated.

The hollow gas piston *J*, and the pistons *O* and *P* in the air pumps, which latter are also provided with automatic valves, are all mounted on one pump rod.

It will be seen from Fig. 2 that the forward end of the gas pump cylinder and the back air pump cylinder must each be in communication with one of the main inlet valves of the power cylinder; and that similarly, the back end of the gas pump cylinder and the forward air-pump cylinder must communicate with the other main inlet valve. This is accomplished by passages cast in the engine bed.

The action of the pumps and power cylinder is as follows:—When the engine piston is on the forward dead centre, as shown in Fig. 2, the exhaust ports *S* are uncovered. At the moment that the pressure in the cylinder falls to that of the atmosphere the admission valve is opened, and a scavenging charge of pure air is sent into the cylinder, thus clearing out the burnt gases. The gas pump follows with a charge of gas into the cylinder, where it mixes with the air previously sent in, and forms the explosive mixture for the power stroke. The exhaust ports *S* are then closed by the return stroke, and the mixture is compressed and ignited, thus forcing the piston outwards. The same cycle is repeated alternately at either end.

The method now adopted for operating the valve gear of the inlet valves is an improvement on the original design, in which cams fitted on a lay shaft were employed, driven from the main crank shaft by skew or bevel gears. These have been eliminated. Referring to Fig. 2, the inlet valves *A* are mounted on spindles *B* passing through the valve chambers, and surrounded by helical springs *C*, which normally hold the valves closed down on their seats. Above each valve spindle is placed a pair of rolling levers *D* and *E* with a nose on the under-side of each lower lever *E*, which bears on the top of the valve spindle, so that when *E* is depressed the valve is opened. As the inlet valves require to be open during about one-quarter only of a revolution of the engine, the surfaces of the levers *D* and *E* are shaped so as to produce a rapid movement of the valve spindle on opening and closing, thus allowing a prolonged pause for the remainder of the revolution. The two pairs of levers are connected by a rod *F* so as to work together with opposite movements, one pair being closed while the other pair is open. They are operated through the connecting-rod *G* and the single eccentric *H* on the crank shaft.

Starting is effected by compressed air admitted through a separate starting valve, and governing is effected on the charging stroke of the gas pump by means of a "Chorlton-Whitehead" pattern governor.

The great regularity of turning is shown in the accompanying reproduction of a "Moscrop" record of a day's run, from which it will be seen that the variation in speed is under one-half per cent.

The Engineer's Book-shelf

Principles of Copper Smelting. By EDWARD DYER PETERS. Pp. vii. + 610. Price $5·00. Hill Publishing Co., New York.

This book is noteworthy for the absence of quotation, for the somewhat meagre description of plant and for its rigid adherence to principles underlying processes rather than to the processes themselves. The processes have been well and fully treated in the author's "Modern Copper Smelting," which has therefore cleared the ground for the present work. The principles are admirably taught, and from the second chapter onwards the book apart from its technical value also gives an admirable course in the process of reasoning. The clearness and simplicity of the opening chapters render the book suitable for practical men in that it takes them by progressive stages through the principles of their work. Technical metallurgists also find this feature of value in that it will gently but surely revivify earlier and possibly forgotten knowledge. To give any idea of the scope of the book is impossible beyond stating that it covers the full range of the metallurgy of copper production and is an exposition of principles brought down to date. A chapter on thermochemistry is from the pen of Prof. Richards—a recognised authority on this subject. In treating Pyrite Smelting the author has had to some extent the co-operation of Mr. R. Sticht and a paper by the latter, "Ueber das Wesen des Pyrit Schmelzverfahrens" in *Metallurgie* of about a year ago has been freely used. Apart from these instances the book is wholly original and remarkably complete. Prof. Peters is to be congratulated on the production of a work which will become classical.

Einführung in die Metallographie. By PAUL GOERENS. Pp. 185. Price 10 marks. Halle a S. Wilhelm Knapp.

Within the compass of a small volume Mr. Goerens gives not only an introduction to the study of metallography, but also presents a clear *resumé* of the present position of this science. Some of the latest work is summarised and dovetailed into earlier results, the combination giving what is almost a narrative of researches published up to about 1905.

Commencing with the physical properties of matter the reader is led to a discussion of cooling curves and their interpretation features which involve a description of methods for measuring and recording temperature. The following section deals with physical chemistry as applied to salts and alloys, a large number of cooling curves are included, and a discussion of them leads up to the constitution of various alloys. The third portion of the book is devoted to the technics of metallography, including the preparation and examination of sections and photography. The last section of the book deals with the metallography of iron and its alloys. The illustrations throughout are telling, and special praise should be given to the photomicrographs.

Open Hearth Steel Castings. By W. M. CARR. Pp. 118. $1·50. The Penton Publishing Co., Cleveland, O., U.S.A.

This work is described as a complete exposition of the methods involved in the manufacture of open hearth steel castings by the basic and acid processes. It is compiled from a series of articles written for the *Iron Trade Review* and the *Foundry*, several of which have been noted in our Iron and Steel section. A perusal of the book leaves one in very considerable doubt as to its completeness, and in particular the chapter on heat treatment and annealing is notoriously incomplete. Not only so but the data given has very little bearing on modern annealing practice. However, the book is well worth the attention of steel founders, and they will find in it much of interest on open hearth practice.

The Chemistry of Gas Manufacture. By Harold M. Royle, F.C.S. London: Crosby Lockwood & Son. Stationers' Hall Court, 1907. 12/6 net.

In this useful manual the author has wisely limited his treatment to certain aspects only of the chemistry of gas manufacture, leaving the operations of manufacture themselves alone. Among the points specially treated may be mentioned coal and its characteristics, furnace gases, products of carbonisation, materials for purification, fire-bricks and fire-clay, photometry and gas testing, and carburetted water gas, all of which require attention in the ordinary course of the gas engineer's duties. Where no separate chemical staff is kept the volume should prove of the greatest help and utility to the engineer and manager: at the same time students will find it a well arranged and up-to-date work.

In this section is presented every month a list of the more important articles appearing in the Engineering Press and of the leading papers read before the various engineering institutions. Full names, addresses, and published prices of the publications indexed are printed each half-year in the volume index.

Civil Engineering.

Reinforced Concrete Slab Bridges. *Railway and Engineering Review*, 18*th January*, 1908.

Structural Steel Work, I. By E. G. Beck, A.M.Inst.C.E. *Mechanical World*, 17*th January*, 1908.

Experimental Investigations of the Stresses in Masonry Dams subjected to Water Pressure. By Sir J. W. Ottley and A. W. Brightmore. *Proceedings Institution Civil Engineers*, 21*st January*, 1908.

Stresses in Dams: An Experimental Investigation by means of Indiarubber Models. By J. S. Wilson and Wm. Gore. *Proceedings Institution Civil Engineers*, 21*st January*, 1908.

The High Needle Dams on the Big Sandy River, U.S.A. *Engineering*, 17*th January*, 1908.

The Flushing Tunnel for the Gowanus Canal in Brooklyn, New York. *Engineering Record*, 11*th January*, 1908.

New Swing Bridge over the River Hull at Sculcoates, Hull. *Engineering*, 31*st January*, 1908.

Sinking Well and Cylinder Foundations. By Ed. W. Stoney. *Engineering*, 31*st January*, 1908.

The New Roof of Charing Cross Station. *Engineering*, 7*th Feburary*, 1908.

Stressesin Solid Beam Sections and the Strength of Chain Rings, I. By R W. Smith. *Engineer*, 24*th January* and 7*th February*, 1908.

The Proportioning of Steel Railway Bridge Members. By H. S. Prichard. *Proceedings Engineers Society, Western Pennsylvania, December*, 1907.

The Erection of the Pwll-y-Pant Viaducts on the Brecon and Merthyr Extension of the Barry Railway. By Alex. Low Dickie. *Transactions Institution Civil Engineers*, 11*th February*, 1908.

Notes on the Erection of Cantilever Bridges. By Prof. T. Claxton Fidler. *Transactions Institution Civil Engineers*, 11*th February*, 1908.

Erection of the Manhattan Bridge across the East River. *Scientific American*, 1*st February*, 1908.

Roofs, VI. *Railway Engineer, February*, 1908.

Safe Stresses in Steel Columns. By J. R. Worcester. *Proceedings American Society Civil Engineers, January*, 1908.

Rivetted Lattice for Railroad Bridges of Maximum Span; a Plea for a Return to Rational Design. By Geo. H. Thomson. *Engineering News*, 23*rd January*, 1908.

The Ohio St. Bascule Bridge at Buffalo, New York. *Engineering News*, 16*th January*, 1908.

Building.

Failures of Reinforced Concrete Chimneys and Recommendations for Design and Construction. By S. E. Thompson. *Engineering News*, 9*th January*, 1908.

The Stadium of Syracuse University. *Engineering Record*, 18*th January*, 1908.

Heating and Ventilation.

Materials and Measurements.

The Analysis of Concrete. By R. L. Wales. *Engineering News*, 9*th January*, 1908.

The Manufacture of Concrete Bricks from Blast Furnace and other Slag. By J. Butler. *Iron and Coal Trades Review*, 24*th January*, 1908.

The Elastic Limit and the Testing of Materials. By H. Gansslen. *Machinery, January*, 1908.

Experience with Concrete in Fortifications. *Engineering Record*, 25*th January*, 1908.

Twelve Tests of Carbon-Steel and Nickel-Steel Columns. *Engineering News*, 16*th January*, 1908.

The Influence of Bond on Size of Reinforcement Bars for Concrete. *Engineering Record*, 25*th January*, 1908.

Boilers, Furnaces, and Fuel.

Smoke Suppression. By E. J. Kunze. *Engineer*, 31*st January*, 1908.

A Recent Plant for the Utilisation of Small Coal. By E. M. Hann. *Colliery Guardian*, 31*st January*, 1908.

The Destruction of Tar in Gas-Producers. By H. P. Bell. *Engineering*, 31*st January* and 7*th February*, 1908.

Engines and Motors.

The Construction and Working of Large Gas Engines. By P. R. Allen. *Proceedings Manchester Association Engineers, 11th January*, 1908.

The Indicator for Valve Setting. *Engineer, Chicago, 1st January*, 1908.

Lining up a Horizontal Engine. Simple and Most Complete Directions for this Work. By S. E. O'Donnell. *Engineer, Chicago, 1st January*, 1908.

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TRADE LITERATURE.

The Warner Engineering Co., Westminster, S.W. Price list of Warner-Solio Arc Lamps for direct and alternating current. **Consolidated Pneumatic Tool Co. Ltd.**, 9, Bridge St., S.W Catalogue No 1. E. of electric tools, comprising drills, grinders, blowers, lifting blocks, and a special type of drill and grinder wound for 500 volt d.-c. traction circuits. **Alfred Herbert, Ltd.,** Coventry. A section of catalogue, 4th Edition, of horizontal and vertical milling machines. Also E section, 16th Edition, of modern capstan lathes. **R. Wolf,** 7, Lawrence Pountney Hill, E.C. Catalogue of portable and semi-portable engines fitted with superheater. **Technical Art Co.,** Newcastle-on-Tyne. Leaflet of winking signs, automatic thermo-electro cone sign and duplex flashing cone sign. **Hopkinson Gas Plant Co.,** Huddersfield. List No. 150 of suction, suction-pressure and pressure gas plants for coke, anthracite, charcoal, &c. **Michael Pal & Co.,** Victoria St., S.W. Catalogue of oil saving and waste-cleaning plants, steam turbine fans, rotary pumps, &c. **Pulsometer Engineering Co.,** Reading. Abridged catalogue of the manufactures of this company comprising pulsometers, centrifugal and turbine, sinking and other pumps, pumping engines, ice making and cooling plants, ice machines, and water-tube boilers. **E. S. Hindley & Sons,** Bourton. Well illustrated catalogue of steam engines, boilers, hoisting and sawing machinery, vertical gas engines, suction gas plants and steam motor vehicles. **Dewhurst's Engineering Co., Ltd.,** Sheffield. List containing description and detail drawings of the G & C hydraulic working valves more particularly adapted for use in steel works; also a well illustrated list of the firm's patent slag ladles and cars for iron and steel works. **Siemens-Schuckert Werke.** Berlin. Pamphlet A.B. 14, "Eine elektrische lokomotiv-förderung für 2,000 Volt gleichstrom, also A.B. 15. Das Umformerwerk "Krumme, strasse" der Berliner elektrischen Hoch-und Untergrundbahn. **The Brooke Tool Manufacturing Co., Ltd.,** Birmingham. Price list of milling cutters made from special carbon and high-speed steels. Other specialties of this firm are machine-relieved form cutters, metal slitting saws, reamers, gear cutters and twist drills.

Mr. Robert Matthews, of Messrs. Sir W. G. Armstrong, Whitworth & Co., Ltd., Openshaw, Manchester, has just been appointed for the second time as the president of the Manchester District Engineering Trades Employers' Association.

From Messrs. Beanland, Perkins & Co., Leeds, we have received a sample box of Gresolvent, a preparation which instantly dissolves and removes machine grease, rust. paint, &c., without injury to the skin. From a trial we can confidently recommend this preparation to machinists, engineers, automobilists, &c.

WORKSHOP PRACTICE PRIZE COMPETITION.

The Fifth Prize Competition announced in the last number closes on March 15th, by which date all papers must be received. This also relates to Workshop Practice, and is limited to Works Managers, Foremen, and those under their charge, to provide an opportunity of bringing their practical ideas and experiences into print.

Four prizes are offered—one of £4 and three of £1—for the best short article describing and illustrating any emergency device or any job which has taxed the ingenuity of the machinist, pattern maker, fitter, smith, or foundry operator.

The award will be announced in the April number.

INSTRUCTIONS AND CONDITIONS.

Articles must be written on one side of the paper only, and be signed by the competitor.

Drawings and sketches must be on separate sheets of paper.

A short biographical sketch of the competitor's technical experience must accompany the article.

The right is reserved of publishing the prize articles, and also any of the others sent in; but in the case of the latter, payment will be made for the same at our usual scale rates.

The Editor's decision in all matters connected with the competitions shall be final, and binding on all the competitors.

All articles should be addressed to the Editor of THE ENGINEERING REVIEW, 104, High Holborn, London, W.C., and marked on the outside, "Competition," and the same must be received not later than March 15th.